GENERAL ENGINEERING LABORATORY

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GENERAL ELECTRIC INDUSTRIAL ACOUSTICS COURSE IV LECTURE NOTES

Design of Filters and Mufflers by D. D. Davis - Langley Aeronautical Laboratory

Case Histories - Noise Reduction by D. C. Apps - General Motors Proving Ground

Report No. R56GL38

March 8, 1956

(NASA-TM-102895) DESIGN OF FILTERS AND MUFFLERS. CASE HISTORIES: NOISE REDUCTION (NASA) 85 p

N90-70581

Unclas

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General Engineering Laboratory

SCHENECTADY, NEW YORK

TECHNICAL INFORMATION SERIES

AUTHOR	SUBJECT CLASSIFICATION	NO.
D. D. Davis D. C. Apps	ACOUSTICS	R56GL38 DATE March 8, 1956

TITLE
GENERAL ELECTRIC INDUSTRIAL ACOUSTICS COURSE IV LECTURE NOTES

ABSTRACT
Lecture notes on the following subjects are contained:
"Design of Filters and Mufflers" - D. D. Davis
"Case Histories - Noise Reduction" - D. C. Apps
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G.E. CLASS		NO. PAGES
I	REPRODUCIBLE COPY FILED AT LIBRARY OF GENERAL ENGINEERING LABORATORY SCHENECTADY, NEW YORK	89
GOV. CLASS. -		

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INFORMATION PREPARED FOR Industrial Acoustics Course IV of the Noise Reduction Program

TESTS MADE BY None

AUTHOR D. D. Davis and D. C. Apps

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ACOUSTIC FILTERS

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Presented in a Seminar at the Acoustic Laboratory of
the General Electric Company

Schenectady, New York
March 8, 1956

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

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INTRODUCTION

The subject of this discussion is the acoustic filter or, more precisely, the use of acoustic filters to reduce noise levels. In the broadest sense, almost any device inserted in a duct or pipe line can be regarded as an acoustic filter. Figure 1 shows a device, labeled "filter" that has been inserted in a duct. In the duct at the left of the filter is an arrow labeled "acoustic input," and at the right is another arrow labeled "output." This discussion concerns filters for which the acoustic output is substantially less than the input over a frequency range that is useful in noise control.

In noise-control work, the need for an acoustic filter usually is encountered when it becomes necessary to transport air or other gases from one place to another through a duct or tube. If one of the requirements of the duct installation is that the transmission of noise through the duct be held to a minimum, then some device must be inserted in the duct to accomplish this. For example, in an air-conditioning system, an acoustic requirement is that the conditioned air be distributed and exhausted without allowing noise that may be generated at blowers, at the conditioning unit, or in flowing through the ducts to enter the air-conditioned rooms. These requirements can be met only by inserting some acoustic attenuating device in the ducting itself.

One of the problems encountered in the installation of internal combustion engines is that of reducing the intake and exhaust noise to acceptable values. This may be accomplished by inserting mufflers in the intake and exhaust ducting to attenuate the pressure pulsations before they reach the surrounding atmosphere.

In some cases, the acoustic problem is not to reduce noise that annoys or endangers people, but to reduce pressure pulsations that endanger engineering structures. In gas pipe lines, for example, if pressure pulsations reach excessive magnitude, the life of the pipe is reduced and the pipe may even fail. The acoustic problem here is to prevent excessive pressure pulsations in the pipe line.

The acoustic problems in the preceding examples are similar in principle. A harmonic analysis of the flow in the duct, in any of these problems, would show pulsating or alternating-current flows at several different frequencies superimposed on an average direct-current flow of gas.

The acoustic problem is to impede the alternating-current flow as much as is practical, while at the same time allowing the direct-current flow to pass. The type of filter that will satisfy these requirements is a low-frequency-pass filter with a cut-off frequency below the lowest alternating-current frequency that is present in the duct. The majority of filter problems that are encountered by noise-control engineers entail the design of this type of filter; hence, this discussion will be limited to low-pass filters. A nonacoustic requirement - that the static pressure drop in the duct not be excessive - often influences the type and size of filter that is employed.

BASIC PRINCIPLES

Two basic principles are used to obtain the filtering action. The first is absorption. This is illustrated in figure 2, where an acoustic input is fed into a device that is capable of absorbing sound energy. A large fraction of the acoustic input energy is converted into heat by the sound-absorbing material in the filter, and only a small part escapes as noise output. Sound-absorbent materials are in general use for reducing high-frequency noise. A typical arrangement is to line the walls of a duct with absorbent material, and this is very effective in the frequency range where the absorption coefficient of the material is high. Figure 3 shows the transmission loss as a function of frequency for a typical duct lined with sound-absorbent material. Note that the transmission loss is greatest in the vicinity of 800 cycles per second. At frequencies below about 200 cycles per second, this lined duct is not very effective in reducing the transmission of noise.

The effectiveness at low frequencies can be increased considerably if the thickness of the absorbent lining is increased, or if an air space is provided between the absorbent material and the outer wall of the duct, as is shown in figure 4. Solid baffles are placed in the air space at intervals along the duct to prevent sound transmission along the air space. This type of structure can be tuned to provide peak transmission loss at a selected frequency by varying such quantities as the air space dimensions and the density and thickness of the layer of acoustic absorbent. The transmission loss curve in figure 4 shows the characteristics of a particular duct of this type (ref. 1). Note that the maximum transmission loss occurs at about 100 cycles per second. Note also that the effectiveness of this tuned duct decreases rapidly with increasing frequency, so that if it is necessary to provide attenuation over a wide range of frequencies, different parts of the duct must be tuned to different frequencies. This is typical of many noise-reduction problems, where different structures, or even different techniques, must often be used in combination to solve a single noise problem. Information useful in the design of filters of this type may be found in references 1 - 5.

The second basic principle that is used in filter design is that of reflection. This is illustrated in figure 5. The sketch at the top of this figure shows a device inserted in series with a duct. If the impedance at the discontinuity between the duct and this device is either much higher or much lower than the characteristic impedance of the duct, there is a serious mismatch at this point. Under these conditions, when a sound wave traveling through the duct arrives at the discontinuity, only a small fraction of the sound energy can be transmitted. The rest of the energy goes into a reflected wave that originates at the discontinuity and travels back down the duct toward the source.

The lower sketch shows a device inserted in parallel with the duct. If the impedance of this device is very high, it will have little effect on the transmission of sound. If, on the other hand, the impedance is much lower than the characteristic impedance of the duct, there will again be a serious mismatch and only a small fraction of the incident sound energy will be transmitted. Consideration of a limiting case will show why noise reduction is obtained with filters of this type. If the impedance of the parallel filter is zero, the filter acts as a short circuit in the duct. All of the acoustic energy must flow into this short circuit; none can be transmitted into the duct beyond the filter.

At this point it may be well to point out that the effectiveness of a filter depends to some extent on what becomes of the reflected wave. This wave is carrying energy back toward the source and it is important to know, for instance, whether the reflected-wave energy will add to the strength of the incident wave. This question will be considered later.

At low frequencies, filters that operate on the principle of reflection are often more effective than absorption filters. On the other hand, when the frequency is high enough so that the sound wave length is of the same order as the filter dimensions, reflection filters are likely to be ineffective. In a particular filter design, it is, of course, possible (and often desirable) to use the principles of reflection and absorption in combination. The relative effectiveness of the two methods of obtaining noise reduction depends on the frequency and the duct dimensions.

The discussion up to this point has been concerned with describing the two basic principles that are used in filter design. This discussion has been rather general in nature and before it can become much more specific, it will be necessary to introduce some method of arriving at a numerical index of the effectiveness of a filter.

INSERTION LOSS AND TRANSMISSION LOSS

The purpose of a filter is to reduce the noise level. Hence, the most direct measure of the effectiveness of a filter would be a comparison of the noise levels with and without the filter in the system. Unfortunately, however, the noise level of a particular sound, which depends on the subjective response of an individual or of a group to that sound, is not a quantity amenable to physical measurement. As a result, the characteristics of a filter are generally described in terms of the sound pressure, which is a quantity that is readily and accurately measurable.

In figure 6, the sketch at the top shows a sound source, a length of duct, and the termination of the duct. This termination might be, for instance, an opening into a room. The lower sketch shows a filter added to this system. If the filter is installed in order to reduce the sound level at the termination, the effectiveness could be determined by comparing the sound pressures p_a, without the filter, and p_b, with the filter. The insertion loss is a quantity that is convenient for making this comparison. It is fundamentally defined, as shown in figure 6, in terms of the logarithm of the pressure ratio. If the pressures are measured with an instrument calibrated in terms of sound pressure level, this definition means that the insertion loss is simply the sound pressure level without the filter, SPL_a, minus the sound pressure level with the filter, SPL_b.

Another quantity that is often used in discussing the characteristics of filters is the transmission loss. This gives the relationship between the energy in the incident wave at the inlet of the filter and the energy in the transmitted wave at the outlet. The definition is given in figure 6. Note that the insertion loss compares pressures at a single point for two systems, one with a filter and the other without a filter; whereas the transmission loss compares pressures at two points for a single system, the one that includes the filter. Although these two quantities are different in general, in one particular case they are not. If the source and terminating impedances are both equal to the characteristic impedance of the duct, then the insertion loss is equal to the transmission loss.

With a means for specifying or describing filter performance at hand, it is now possible to consider the problem of filter design in some detail.

FILTER DESIGN PROBLEM

The problem that is faced by the filter designer may be conveniently divided into three parts. The first is to formulate the design requirements in terms of engineering units. Normally this will involve the specification of a curve of insertion loss or transmission loss plotted against frequency. The filter performance must either meet or exceed this curve throughout the frequency range. There is no intention to discuss this part of the problem in the present lecture. The second part of the filter designer's problem is to arrive at the physical configuration and dimensions for a filter that is intended to meet the specified requirements. The third part of the problem is to evaluate the performance of the filter experimentally in order to determine whether it is satisfactory.

Consider the second part of this problem; that is, given a curve of required insertion loss plotted against frequency, how can a filter be designed to meet the requirements? To do this without resort to an excessive amount of cut-and-try experimentation, the designer needs to know certain things. Most of all, he must know how to compute or predict the performance of a given filter. Furthermore, it is helpful if he has available some systematic design approach that will shorten the time required to arrive at approximate filter dimensions by eliminating the necessity for involved computations.

Before presenting any design curves, it will be instructive to consider in full the problem of computing the performance of a filter. The procedure that is followed in making such a computation will be illustrated for a particular case.

INSERTION LOSS FOR A SINGLE-CHAMBER PARALLEL FILTER

Figure 7 shows first an equivalent circuit for the source-duct-termination system at the top of figure 6. Below this is the equivalent circuit for this system with a filter installed. This circuit applies to a single-chamber filter installed in parallel with the duct. The filter impedance is Z_f. In order to determine the insertion loss, it is necessary first to compute the pressures p_a and p_b. This computation can be handled by the usual methods of electrical or acoustical circuit theory. The duct sections cannot be treated as lumped impedances except at very low frequencies. Generally, it is necessary to treat the duct as a transmission line of characteristic impedance Z_o. The insertion loss is given by 20 times the logarithm of the absolute value of the ratio p_a/p_b, resulting in the equation shown at the bottom of figure 7. The insertion loss is found to depend on the following parameters:

1. Filter impedance
2. Source impedance
3. Termination impedance
4. Duct lengths l_1, l_2

Impedance diagrams will be presented to illustrate the effects of these parameters.

Figure 8 is a plot illustrating the variation of the magnitude of the impedance for a source-duct-filter system at a particular frequency. At the right is the termination, with a low impedance. Moving to the left, the impedance increases to a very high value just at the right of the filter, which is nearly a quarter of a wave length from the termination. The filter impedance is quite low, so the impedance in the duct to the left of the filter is low. Moving further to the left, the impedance rises until it is well over twice the characteristic impedance, Z_o, at the source.

Because the filter impedance Z_f is so much less than the impedance to the right of the filter, most of the sound energy will flow into the low-impedance filter. If the filter impedance is increased, the filter becomes less effective. It is apparent that the filter impedance has a strong influence on the insertion loss, and that high insertion loss requires low filter impedance.

The source impedance Z_s determines what becomes of the reflected wave in the inlet duct. If $Z_s = Z_o$, then the reflected wave is absorbed by the source and it does not affect the strength of the incident wave. Higher or lower values of Z_s will result in partial re-reflection of the reflected wave, which may, depending on the phase, either increase or decrease the strength of the incident wave. Referring to figure 8, the maximum sound energy will be transmitted when the source impedance is exactly matched to the impedance seen in the duct at the source. The insertion loss can be increased by creating a serious mismatch at the source.

The effect of a change in the termination impedance Z_t is shown in figure 9. The impedance diagram from figure 8 has been reproduced at the top. If the termination impedance is very high, the lower impedance diagram is obtained. Here the impedance at the right of the filter is lower than Z_f. As a result, the filter will be much less effective than in the system at the top of figure 9.

The effect of a change in the length of duct between the filter and the termination is shown in figure 10. In the lower diagram, the termination is nearly half of a wave length from the filter. The duct impedance at the right of the filter is greatly reduced, and the filter becomes ineffective. Note that the same effect has been produced in two ways, first by increasing the termination impedance and then by increasing the duct length. As a further illustration of the interrelationship between these two parameters, note that if, in the lower system of figure 10, the termination impedance were made very high, the filter would again become highly effective.

The effect of a change in the length of duct between the source and the filter is shown in figure 11. Increasing the length of this duct causes a reduction in the duct impedance at the source. Thus, a source that was matched to the upper system would not be matched to the lower system.

In one special case, this problem is greatly simplified. If the source and termination impedances are both Z_o, the duct lengths have no effect on the insertion loss. Furthermore, the insertion loss is equal to the transmission loss. The equation is (ref. 6)

$$TL = 20 \log_{10}\left|1 + \frac{Z_o}{2Z_f}\right| db$$

This equation is used in the study of single-chamber parallel filters, and it can be used to obtain design curves for such filters. To illustrate how this equation is used, a particular type of filter will be considered.

DESIGN OF SINGLE-CHAMBER RESONATOR FILTERS

Figure 12 shows a schematic sketch of a single resonator filter (ref. 6). The filter impedance, neglecting resistance, is given as Z_r under the sketch. The volume chamber surrounding the duct acts in a manner analogous to a capacitative reactance in an electrical circuit. The short tube opening into this chamber is analogous to an inductive reactance in an electrical circuit. The inductive reactance is $\omega\rho_o/c_o$, where ω is the circular frequency, ρ_o is the mean fluid density, and c_o is the conductivity of the opening. The conductivity is a function of the length and diameter of the connecting tube. The capacitative reactance is $\rho_o c^2/\omega V$, where c is the velocity of sound in the fluid, and V is the volume of the chamber. Substituting these impedances in the transmission loss equation, and with some algebraic manipulation, the equation can be brought into the form shown in figure 12. Here f_r signifies the resonant frequency; that is, the frequency at which $Z_r = 0$. In this form, the equation shows that the transmission loss can be described as a function of only two parameters, $\sqrt{c_o V}/2S$ and f/f_r.

A set of transmission loss curves, calculated for several values of $\sqrt{c_o V}/2S$, are shown in figure 13 (from ref. 6). As $\sqrt{c_o V}/2S$ increases, the transmission loss increases. For this reason, the parameter $\sqrt{c_o V}/2S$ has been called the attenuation parameter.

Figure 14 (data from ref. 6) has been included at this point in order to illustrate the accuracy with which the equation predicts the transmission loss of an acoustic filter of this type. The filter is shown at the left. The solid curve is the calculated transmission loss, the circles represent experimentally measured values. The calculated infinity at the resonant frequency occurs only because the resistance has been neglected, and, of course, is not found in the experiment. The resistance is quite small, however, and the attenuation at resonance is high.

The effect of adding resistance, for example in the connecting tube, is shown in figure 15. A large drop in the transmission loss is found near resonance, but there is some increase at much lower or higher frequencies.

Curves such as those shown in figures 13 and 15 are useful in the design of resonator filters. Assume that the filter requirements are specified by a curve of insertion loss plotted against frequency. By assuming a value for f_r, these requirements can be drawn on figure 13. If a resonator curve for a particular value of the attenuation parameter completely envelops this specified curve, that resonator will satisfy the requirements. The resonator volume can then be computed as

$$V = \frac{\sqrt{c_o V}}{2S} \times \frac{cS}{\pi f_r}$$

The filter volume can be minimized by choosing several values for f_r, finding the required values of the attenuation parameter, computing the volumes, and selecting the smallest.

The conductivity is given by

$$c_o = \left(\frac{2\pi f_r}{c}\right)^2 V$$

In designing the opening or openings to provide this conductivity, it is important to make the size of the opening large enough so that the particle displacement will be small, of the order of the length of the opening. An approximate equation that relates the conductivity of an opening to the dimensions is

$$c_o = \frac{S_c}{1 + 0.8\sqrt{S_c}}$$

where S_c is the area and l is the length of the opening.

By using a procedure such as the one that has been outlined, it is possible to arrive at resonator dimensions rather rapidly. The next step in the design process is to compute the insertion loss of the resonator in the system with which it will be used. This computation brings in the effects of the actual source and termination impedances and the duct lengths. Normally some deficiency will be found, and either the dimensions or the location of the resonator may be altered to correct the deficiency.

As a practical matter, it may be difficult to find information on the source impedance for various types of sources. It would help the filter designer if experimentalists would measure and publish the pressure and impedance characteristics of various types of noise sources whenever the opportunity presents itself.

Of course, there are limitations on the range of validity of the resonator equation. For example, if the chamber length or diameter approaches one-half wave length, this equation does not apply. Thus, there are practical limits on the chamber volume. If the design curves indicate the need for an impractically large volume, then a single resonator will not be satisfactory.

FILTERS WITH MORE THAN ONE RESONATOR

When it is found that a single resonator will not provide enough transmission loss to satisfy the requirements, an obvious alternative is to place two identical resonators in the duct, one behind the other. Figure 16 shows the transmission loss characteristics of such a filter (ref. 6). By adding additional resonators, the transmission loss can be increased substantially, but at the same time certain pass bands are introduced. These pass bands should not be allowed to fall in the frequency range where transmission loss is required. If they do fall in that range, additional filter sections will be required to cover the pass bands.

Frequently, the width of the stop band for a single resonator will be found to be insufficient. In this case, the solution is to add one or more additional chambers, each of which is tuned to a different frequency. The combination, if properly designed, will have a much wider stop band than a single resonator. Figure 17 shows the transmission loss, as calculated and as measured, for a filter of this type (ref. 6). The experimental results show a transmission loss of more than 10 decibels over a wide frequency band.

There are several ways to treat combinations of resonators, but in any case the computations are rather time-consuming. One such method (ref. 7) will be described. Figure 18 shows a filter with a reflection-free (Z_o) termination. Two quantities are defined in this figure. The ratio of the reflected pressure to the incident pressure is called the characteristic reflection factor, R_{01}. The ratio of the transmitted pressure to the incident pressure is called the characteristic transmission factor, T_{01}. These characteristic reflection and transmission factors can be computed from the equations for a filter with Z_o termination.

Figure 19 shows this same filter with a reflecting termination. The reflection and transmission factors for the filter installed in this system are given by the equations below the sketch. Note that they can be computed from the characteristic factors, the reflection factor of the termination, and the length of duct between the filter and the termination. The symbols T_{02} and R_{02} refer to characteristic transmission and reflection factors for sound entering the filter from end 2. If the filter is symmetrical, $R_{01} = R_{02}$ and $T_{01} = T_{02}$.

If the transmission loss for a filter of several sections is to be computed, the procedure is to start with the last section and apply these equations. The next step is to move forward to the next section and apply the equations again, using the reflection factor of the last section as R_T. By repeated application of these equations, the transmission factor of each section is determined in turn. Finally, the transmission factor of the complete filter is the product of the transmission factors of the individual sections. Thus, this method provides a systematic means of calculating the transmission loss of multielement filters.

EFFECTS OF OPERATING CONDITIONS

The effects of what might be called operating conditions on the performance of resonator filters are of practical interest. Temperature effects will be considered first. In the equations for transmission loss or resonant frequency, the speed of sound appears. The number that must be used in this connection is the speed of sound in the duct at the operating temperature. The resonant frequency is directly proportional to the speed of sound. In some cases, the chamber may be at a lower temperature than the connector and the duct. In these cases, the speed of sound at the temperature of the duct and connector should be used.

Early in this discussion, the point was made that the need for a filter generally arises from the necessity for transporting air or other gases from one place to another through a duct. Thus, there is, in addition to the sound flow, a steady gas flow to be considered. The effects of this steady flow on the transmission loss of resonators were investigated in a model test in connection with the acoustical treatment of a wind tunnel (ref. 1). Smooth air flow was found to have no appreciable effect on the transmission loss, but turbulent air flow was found to reduce the transmission loss. Larger effects might be expected if the duct flow velocity approached the speed of sound.

Another item that comes under the general heading of operating conditions is the sound pressure level in the duct. The theory that has been used is linear, and in this theory, the transmission loss is independent of the sound pressure level. In deriving the theory, the acoustic pressures and particle displacements are assumed to be small. If they are not sufficiently small, certain nonlinear effects will be encountered, with the result that the theory will not apply. The most critical location in a resonator filter is usually the orifice or tube that connects the chamber with the duct. The area of the connector must be sufficiently large so that transmission of the necessary sound current does not require excessive particle displacements.

FILTER APPLICATION

An example will now be discussed in which a filter was designed to reduce the noise radiated from a helicopter. This example is taken from reference 6. Figure 20 is a photograph of a helicopter powered by a 185-horsepower radial engine. The small tail rotor had been removed at the time this photograph was taken. Noise measurements indicated that the engine-exhaust noise from this helicopter was louder than the noise from any other source. It was expected, therefore, that a significant reduction in overall noise might be obtained simply by installing engine exhaust mufflers. The engine exhaust was carried by two exhaust pipes, one on each side of the fuselage. Hence, two mufflers, one for each side, were installed. This photograph was taken with the mufflers installed.

Figure 21 shows a frequency analysis of the noise from this helicopter with no mufflers. The sound pressure level is highest in the vicinity of the engine firing frequency, as is normal for internal-combustion engines. Four different mufflers were designed for this engine, but only one will be discussed here. Figure 22 shows a sketch of this muffler. It is composed of an annular chamber surrounding the exhaust pipe, with seven holes providing the connection between the pipe and the chamber. It is thus recognizable as a single resonator, terminated by a length of pipe that is open at the end.

The insertion loss for this muffler was measured before it was placed on the helicopter. Figure 23 shows a schematic diagram of the apparatus that was used for making these measurements. Sound pressure levels at a distance of 20 inches from the open end of the muffler were compared with levels at the same distance from the end of the open pipe. An interesting feature of the apparatus is the two-position valve that permits rapid switching from the muffler to the open pipe. The experimental points that were measured with this apparatus are compared with the calculated curve in figure 24. This comparison shows that the performance of an acoustic filter in an actual installation can be predicted with accuracy sufficient for design purposes.

This particular filter was designed for another type of installation, however; installation on a helicopter. Figure 25 shows the left-hand muffler as it was actually installed on the helicopter. Figure 26 shows the effect of the mufflers on the sound pressure level at a distance of about 200 feet from the helicopter. The sound level has been reduced substantially in the frequency range for which the muffler is designed. Note that the reduction measured here cannot properly be compared with the calculated or measured insertion loss of the muffler because of the presence of noise from other sources. The data of figure 26, for example, include an unknown amount of engine intake noise at the same frequencies as the exhaust noise. The overall sound pressure level is reduced 6 decibels by the mufflers. The loudness of the noise, as perceived by the average human ear, is reduced by about 45 percent (based on ASA Standard Z24.2 - 1942).

The second example of the practical application of filters will be drawn from an entirely different field. The operation of wind tunnels can generate rather high noise levels, especially where all or part of the air in the tunnel circuit is exhausted to the atmosphere. At the Lewis Flight Propulsion Laboratory of the NACA at Cleveland, a problem of this type was encountered with a supersonic tunnel. The solution of this problem required the use of many different techniques because the sound level was excessive throughout the frequency range from about 5 cps up to 10,000 cps or more. The sound treatment that was adopted is shown in figure 27 (ref. 1). Resonators, tuned ducts, parallel baffles, and right-angle bends were all used in this system. Figure 28 summarizes the effect of the treatment on the noise measured near the baffles where the exhaust air leaves the tunnel. The noise measured after the sound treatment was installed was found to be coming largely from the surrounding countryside rather than from the wind tunnel.

EXPERIMENTAL EVALUATION OF FILTERS

After a filter has been designed and constructed, the next step is to evaluate the performance of the filter experimentally in order to determine whether it is satisfactory. In the preceding section, an apparatus was described that is suitable for evaluating the performance of a particular type of duct-filter system. In this section, the use of another type of apparatus will be described.

A sketch of this apparatus, which was originally described in reference 8, is shown in figure 29. The filter in this case is simply a right-angle bend in a tube of square cross section. A loudspeaker feeds sound into the input tube. The output tube is terminated with a highly absorbent cone in order to eliminate reflections. A movable probe microphone is used to measure the variation of sound pressure with distance in the input and output tubes. The phase relationships can also be measured if desired.

A sample of the data obtained with this apparatus is presented in figure 30. The upper curve shows the variation of sound pressure with distance in the input tube. Pressure minima occur at intervals of one-half wave length. From this plot, the ratio $P_{1_{min}}/P_{1_{max}}$ is determined. The lower curve shows the variation of sound pressure with distance in the output tube. The plotted quantity is the ratio of the pressure measured in the output tube to $P_{1_{max}}$ in the input tube. At some distance from the filter, this ratio becomes constant, and that constant value is called $P_{1_{tr}}/P_{1_{max}}$. Knowing $P_{1_{tr}}/P_{1_{max}}$ and $P_{1_{min}}/P_{1_{max}}$, the magnitudes of the reflection and transmission factors for the filter can be computed from the equations shown at the bottom of figure 30. Because the filter is terminated without reflection, these are characteristic reflection and transmission factors. The transmission loss is given by

$$TL = 20 \log_{10} \frac{1}{|T|} \text{ db}$$

In the case of a dissipationless filter, the experimental measurement can be simplified. A fixed measuring station in the output tube will give $P_{1_{tr}}$, if it is at a distance of one-half wave length or greater from the output plane. The probe in the input tube need have only sufficient range of movement to measure $P_{1_{max}}$. The transmission loss will then be given by

$$TL = 20 \log_{10}\left[\frac{1}{2}\left(\frac{P_{1_{tr}}}{P_{1_{max}}} + \frac{P_{1_{max}}}{P_{1_{tr}}}\right)\right] \text{db}$$

This simplification is made possible by the fact that the sum of the squares of the magnitudes of the reflection and transmission factors is equal to one for dissipationless filters.

CONCLUDING REMARKS

In summary, this discussion of filter design has included a description of the physical principles that give rise to a filtering action. The physical quantities that are used to describe the performance of a filter have been defined. The problem of designing a filter has been discussed for the particular case of resonator filters. Certain experimental methods for evaluating the performance of filters have been described. Finally, experimental data have been presented to demonstrate that the transmission loss of resonator filters can be predicted by calculation.

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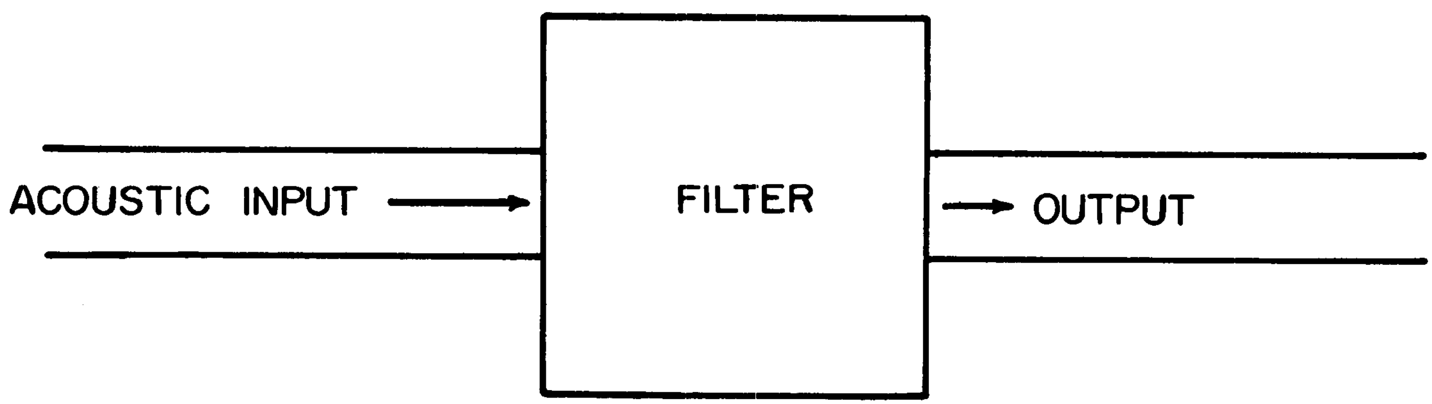
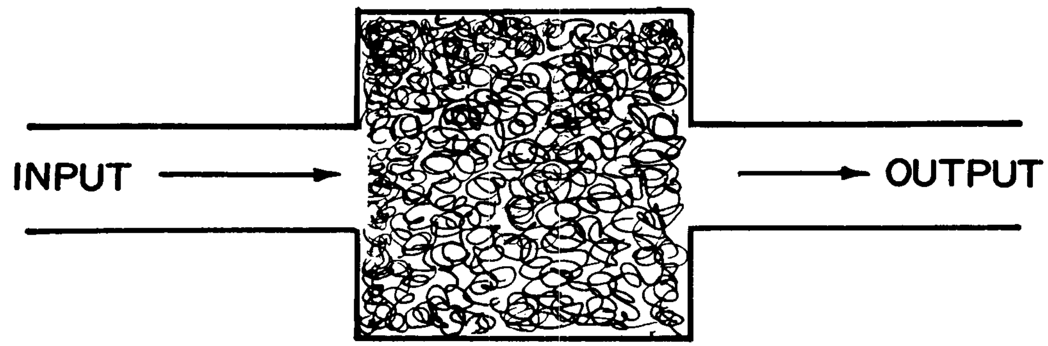
8. Lippert, W. K. R.: A Method of Measuring Discontinuity Effects in Ducts. Acustica, Vol. 4, pp. 307-312, 1954.

DUCT WITH FILTER
(SCHEMATIC)



NACA L-975-1 DAVIS 3/8/56

ABSORPTION FILTER
(SCHEMATIC)



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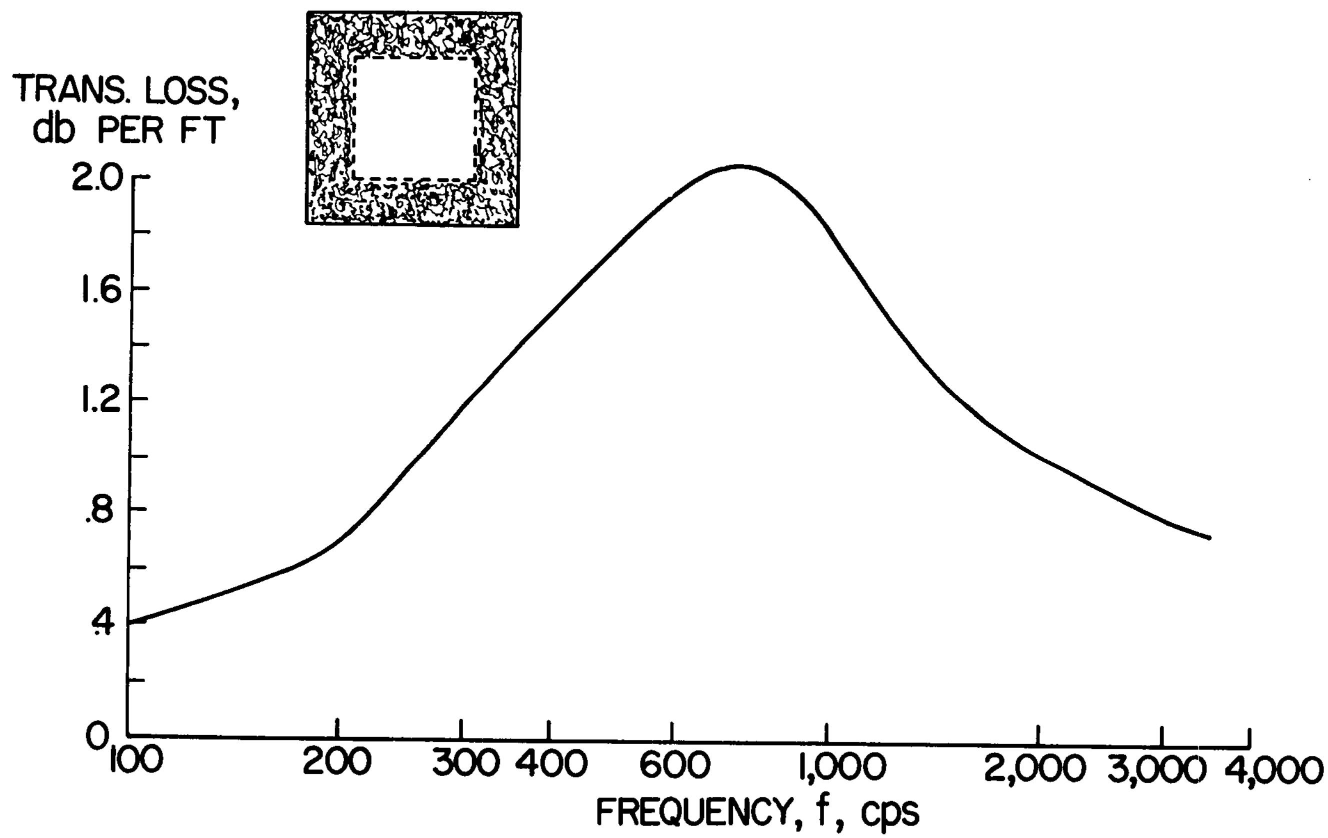
L-975-2 DAVIS 3/8/56

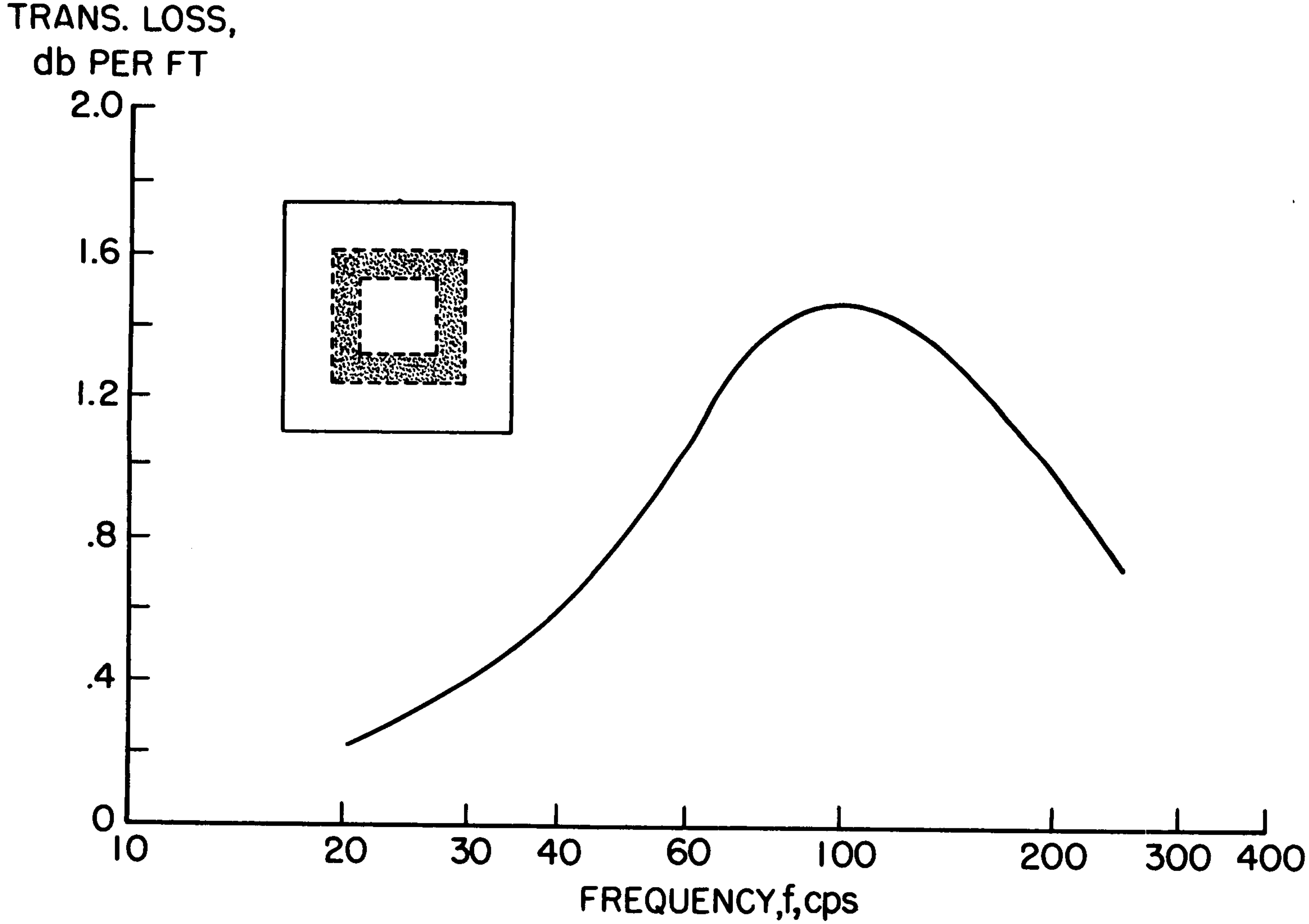
TRANSMISSION LOSS-LINED DUCT



L-975-3

DAVIS 3/8/56

TRANSMISSION LOSS – LINED DUCT WITH AIRSPACE



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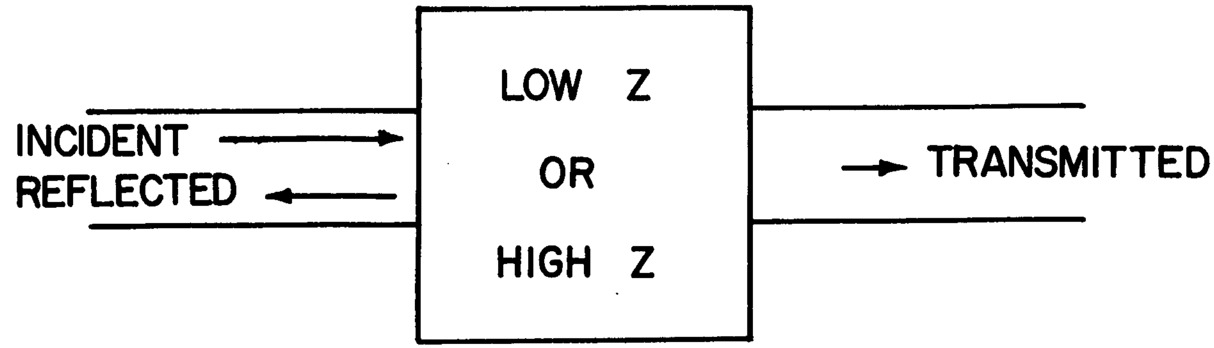
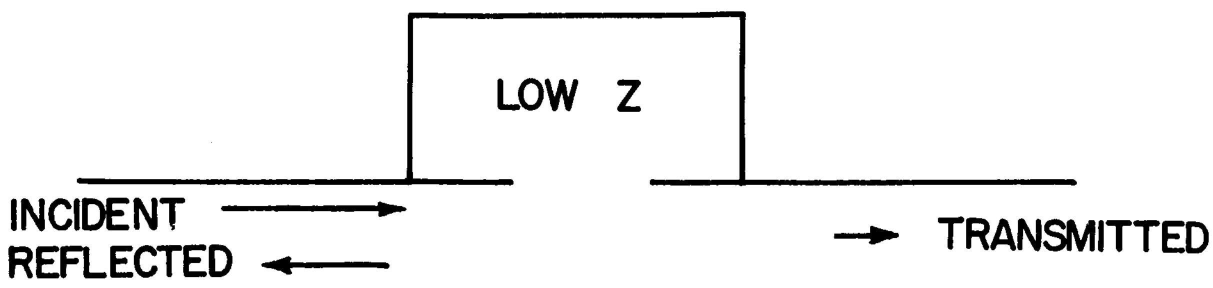
REFLECTION FILTERS
(SCHEMATIC)



SERIES FILTER



PARALLEL FILTER

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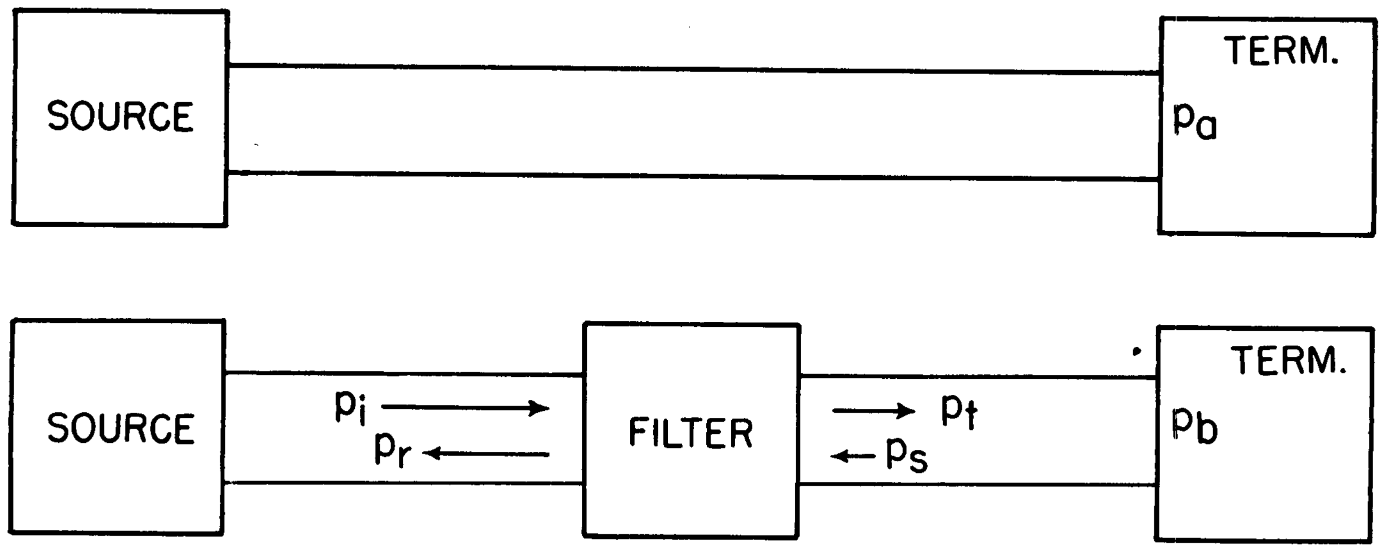
DEFINITIONS - INSERTION LOSS AND TRANSMISSION LOSS



$$IL = 10 \; LOG_{10} \frac{|p_a|^2}{|p_b|^2} = SPL_a - SPL_b \; db$$

$$TL = 10 \; LOG_{10} \frac{|p_i|^2}{|p_t|^2} \; db$$

WHEN $Z_{SOURCE} = Z_{TERM.} = Z_O$

THEN $IL = TL$

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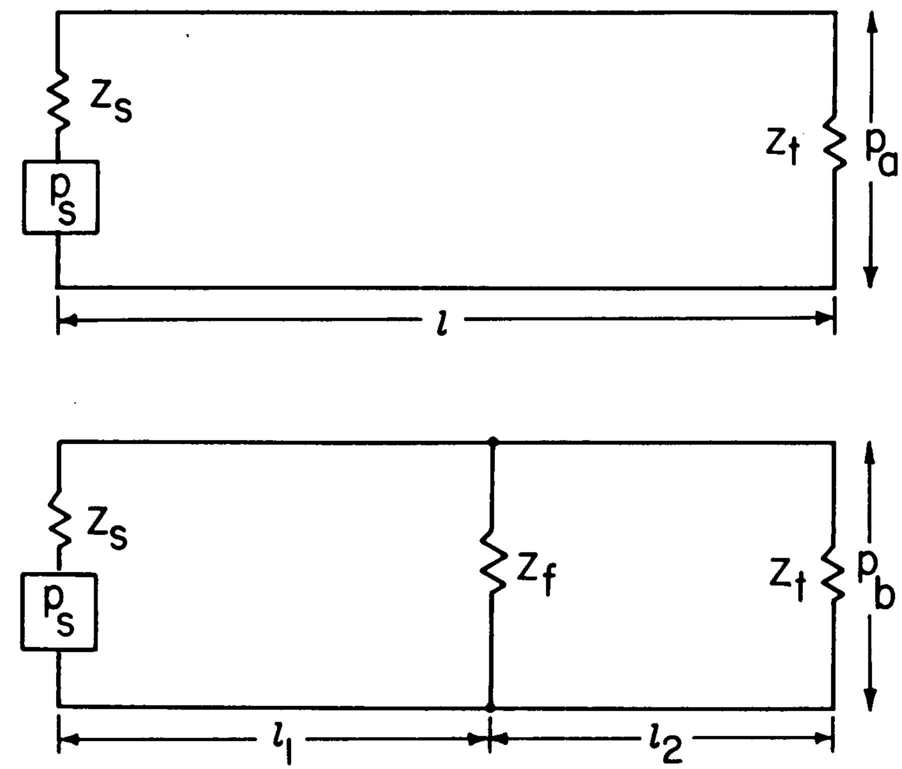
DUCT WITH PARALLEL FILTER

EQUIVALENT CIRCUITS



$$IL = 20 \log_{10} \left| 1 + \frac{\frac{Z_s}{Z_f}\left(\cos kl_1 + i\frac{Z_o}{Z_s}\sin kl_1\right)\left(\cos kl_2 + i\frac{Z_o}{Z_t}\sin kl_2\right)}{\cos kl\left(1+\frac{Z_s}{Z_t}\right) + i\sin kl\left(\frac{Z_o}{Z_t}+\frac{Z_s}{Z_o}\right)} \right| \text{db}$$

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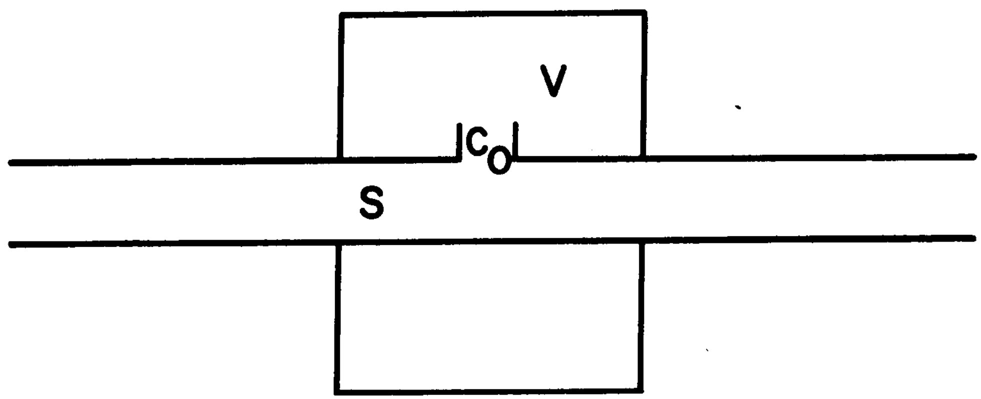
SINGLE RESONATOR



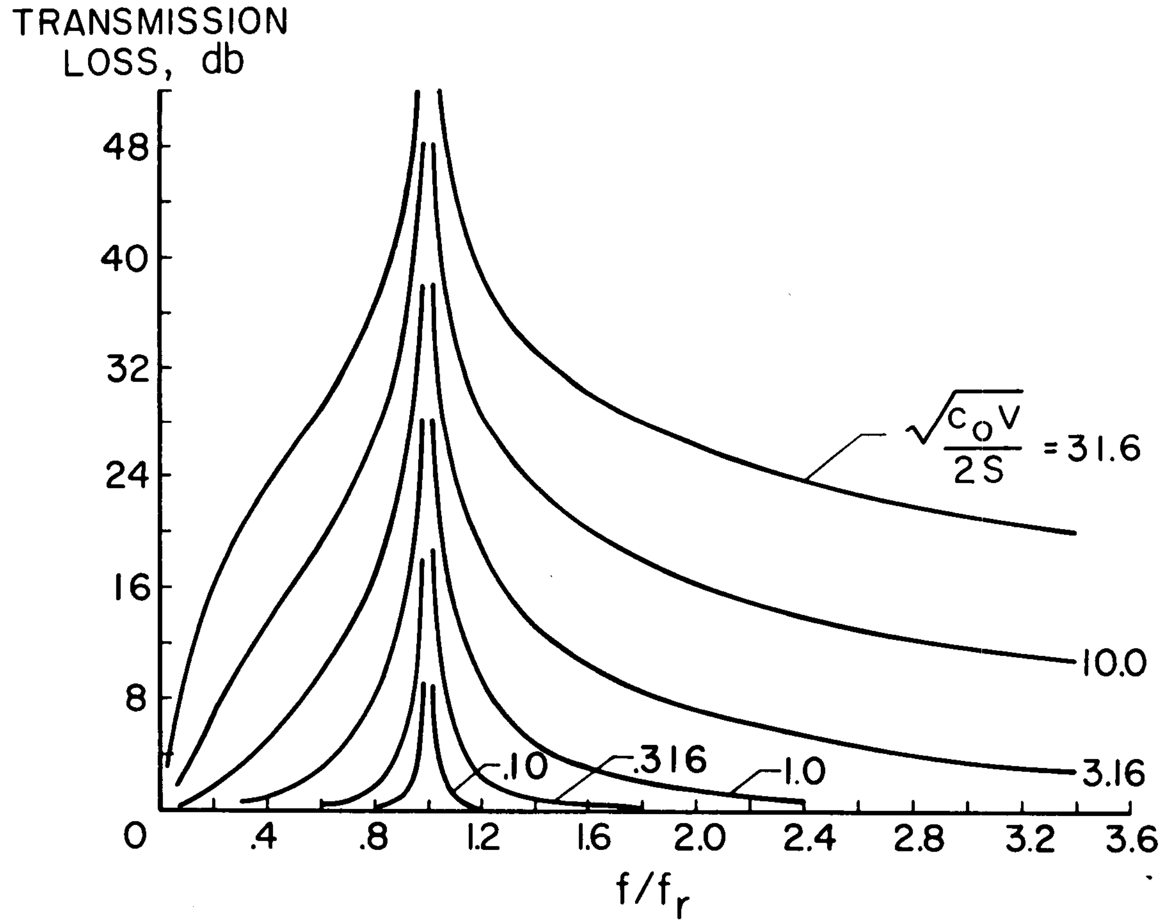
$$Z_r = i\left(\frac{\omega\rho_o}{c_o} - \frac{\rho_o c^2}{\omega V}\right)$$

$$TL = 10\,LOG_{10}\left[1 + \left(\frac{\frac{\sqrt{c_o V}}{2S}}{\frac{f}{f_r} - \frac{f_r}{f}}\right)^2\right] \text{ db}$$

L-975-8 DAVIS 3/8/56

TRANSMISSION LOSS—SINGLE RESONATOR

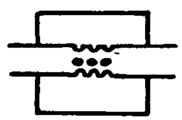
ZERO RESISTANCE



L-975-9 DAVIS 3/8/56

EXPERIMENTAL DATA

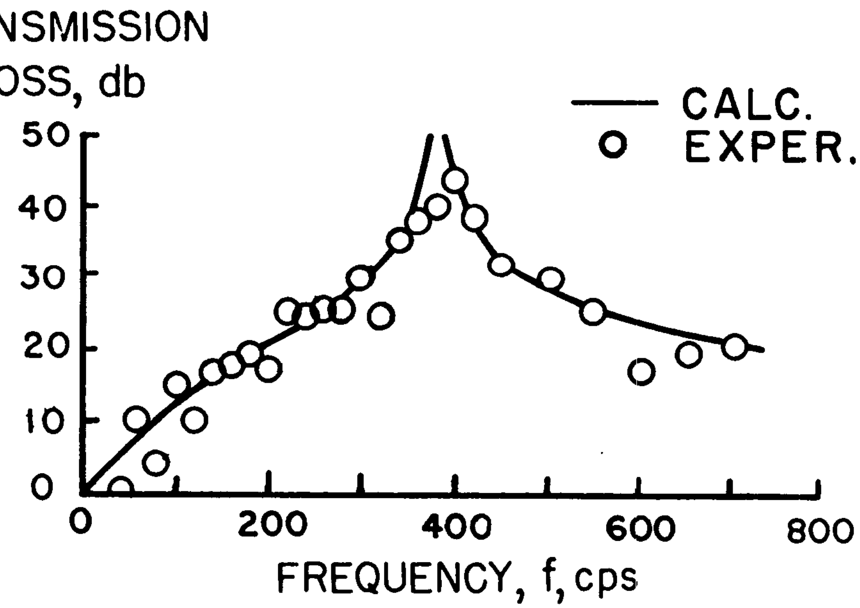
SINGLE RESONATOR



$c_0 = 2.78$

$V = 0.638$

$$\frac{\sqrt{c_0 V}}{2S} = 13.3$$



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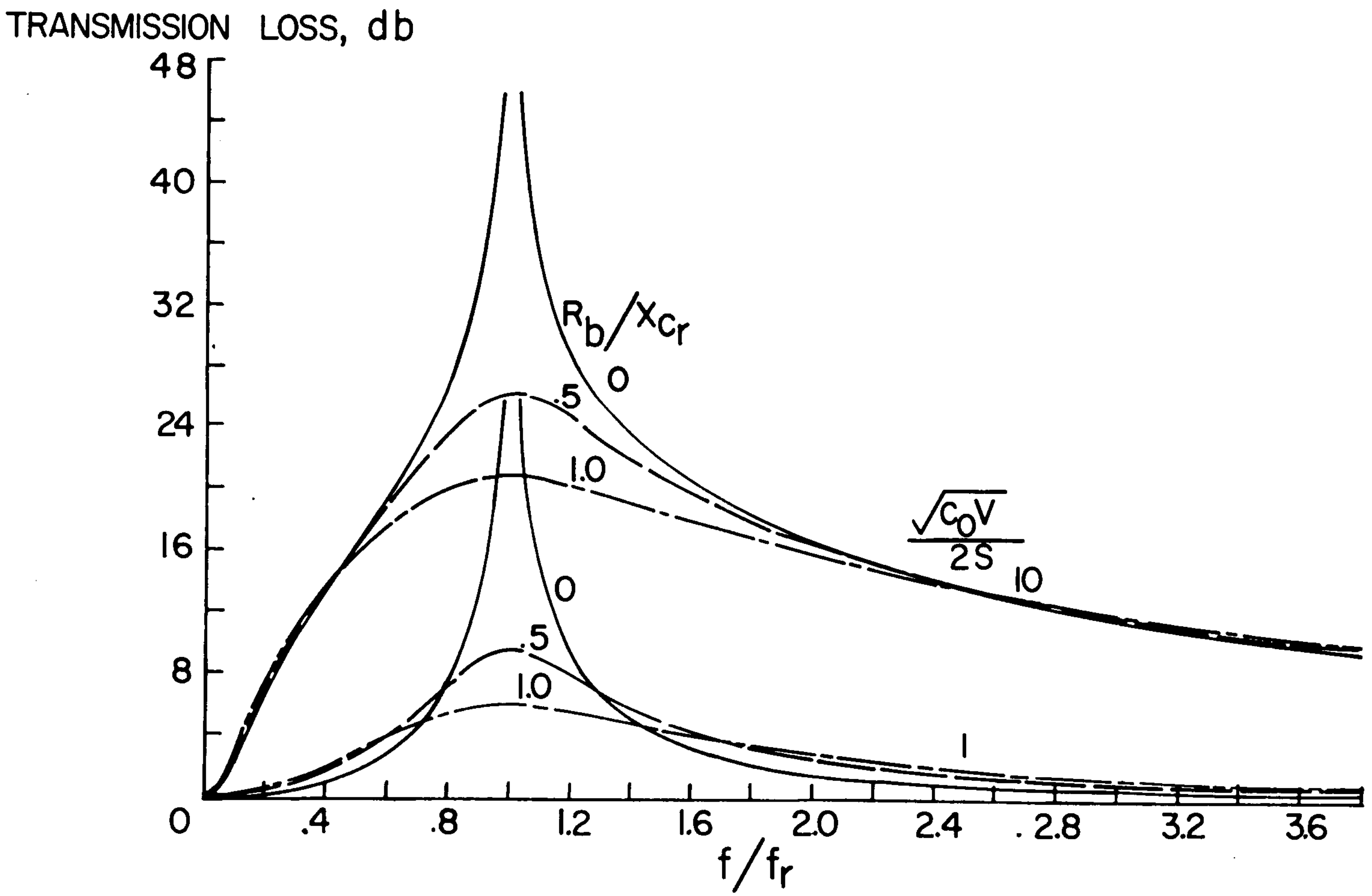
EFFECT OF RESISTANCE

SINGLE RESONATOR



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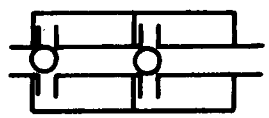
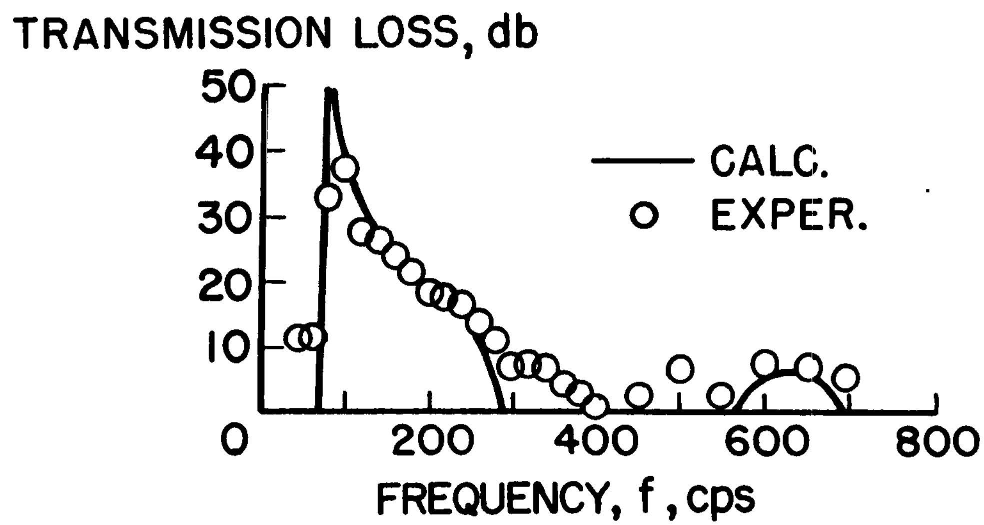
EXPERIMENTAL DATA
DOUBLE RESONATOR



$C_O = 0.14$

$V = 0.736$



$$TL = 8.69\ m\ \cosh^{-1}\left|\cos\left(k_r l_1 \frac{f}{f_r}\right) + \frac{\frac{\sqrt{c_O V}}{2S}}{\frac{f}{f_r} - \frac{f_r}{f}} \sin\left(k_r l_1 \frac{f}{f_r}\right)\right| \text{db}$$

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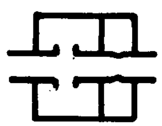
EXPERIMENTAL DATA

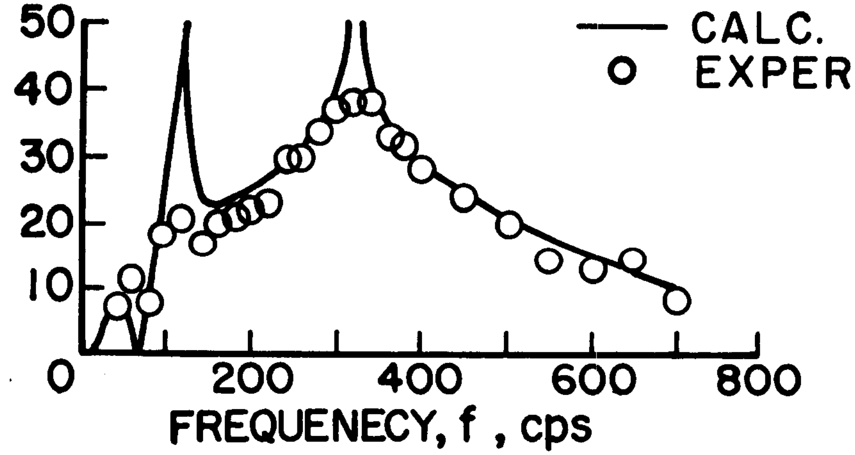
STAGGERED RESONATOR

TRANSMISSION LOSS, db



$C_{O_2} = 0.236$ $\quad C_{O_5} = 0.840$

$V_2 = 0.54$ $\quad V_5 = 0.27$



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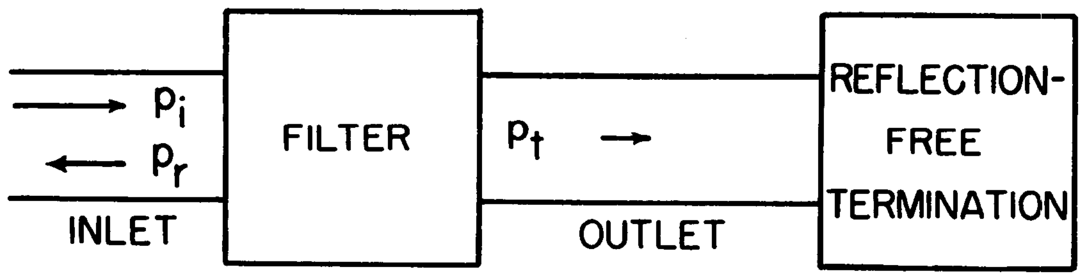
CHARACTERISTIC REFLECTION AND TRANSMISSION FACTORS

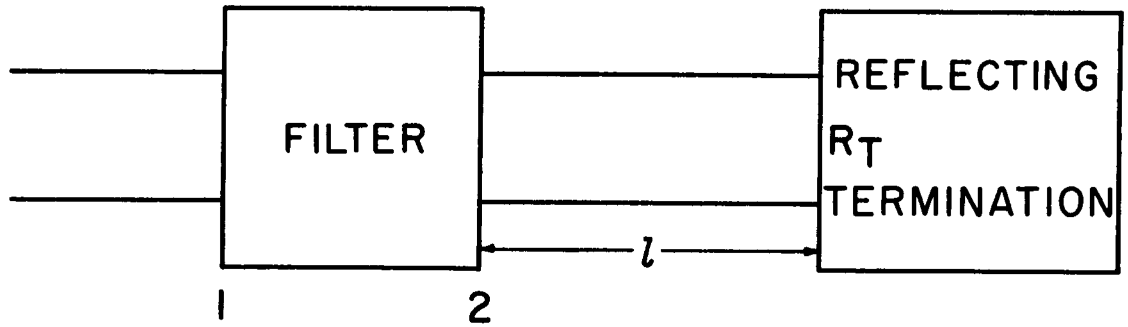


$$R_{OI} = \frac{p_r}{p_i}$$

$$T_{OI} = \frac{p_t}{p_i}$$

L-975-14 DAVIS 3/8/56

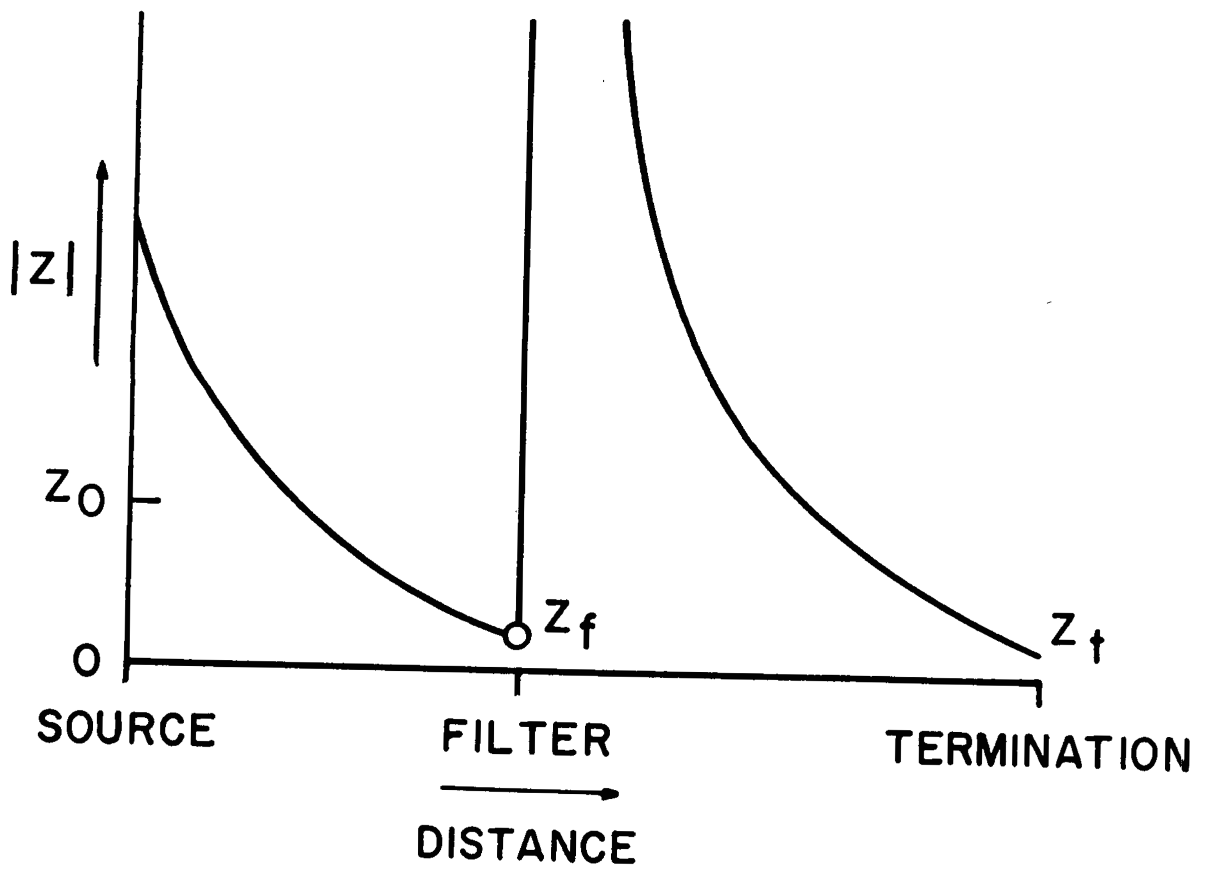
REFLECTION AND TRANSMISSION FACTORS



$$R_1 = R_{01} + \frac{R_T T_{01} T_{02} e^{-ik2l}}{1 - R_T R_{02} e^{-ik2l}}$$

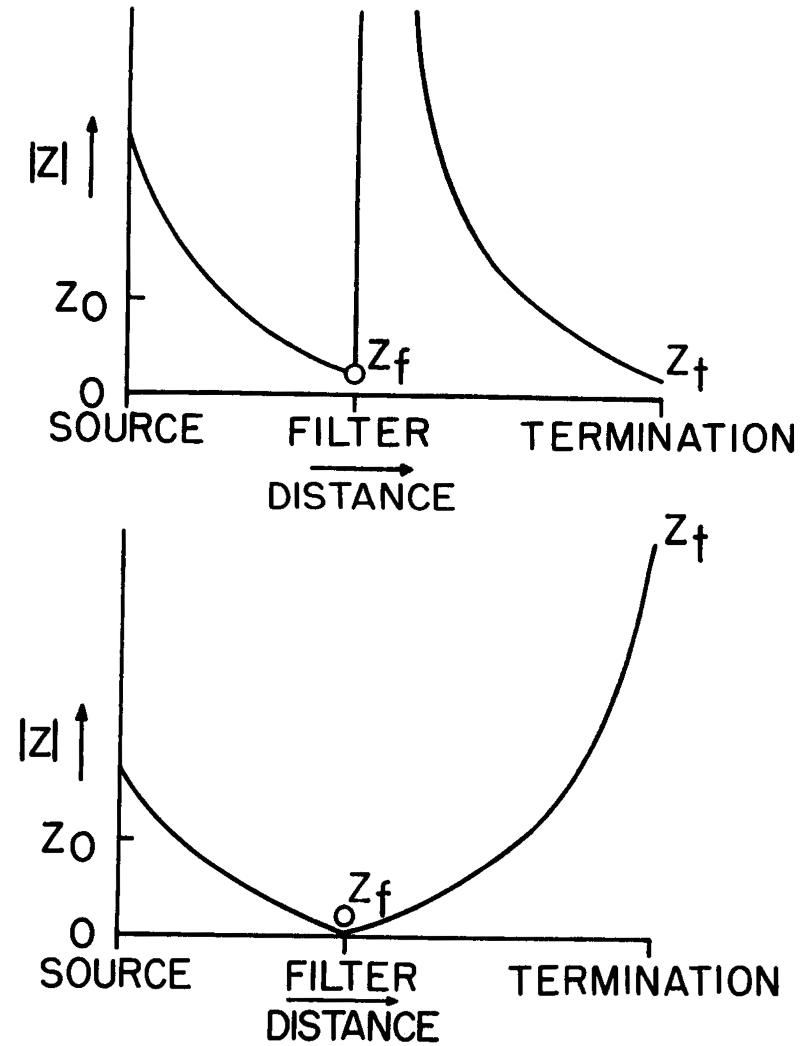
$$T_1 = \frac{T_{01}}{1 - R_T R_{02} e^{-ik2l}}$$

IMPEDANCE DIAGRAM



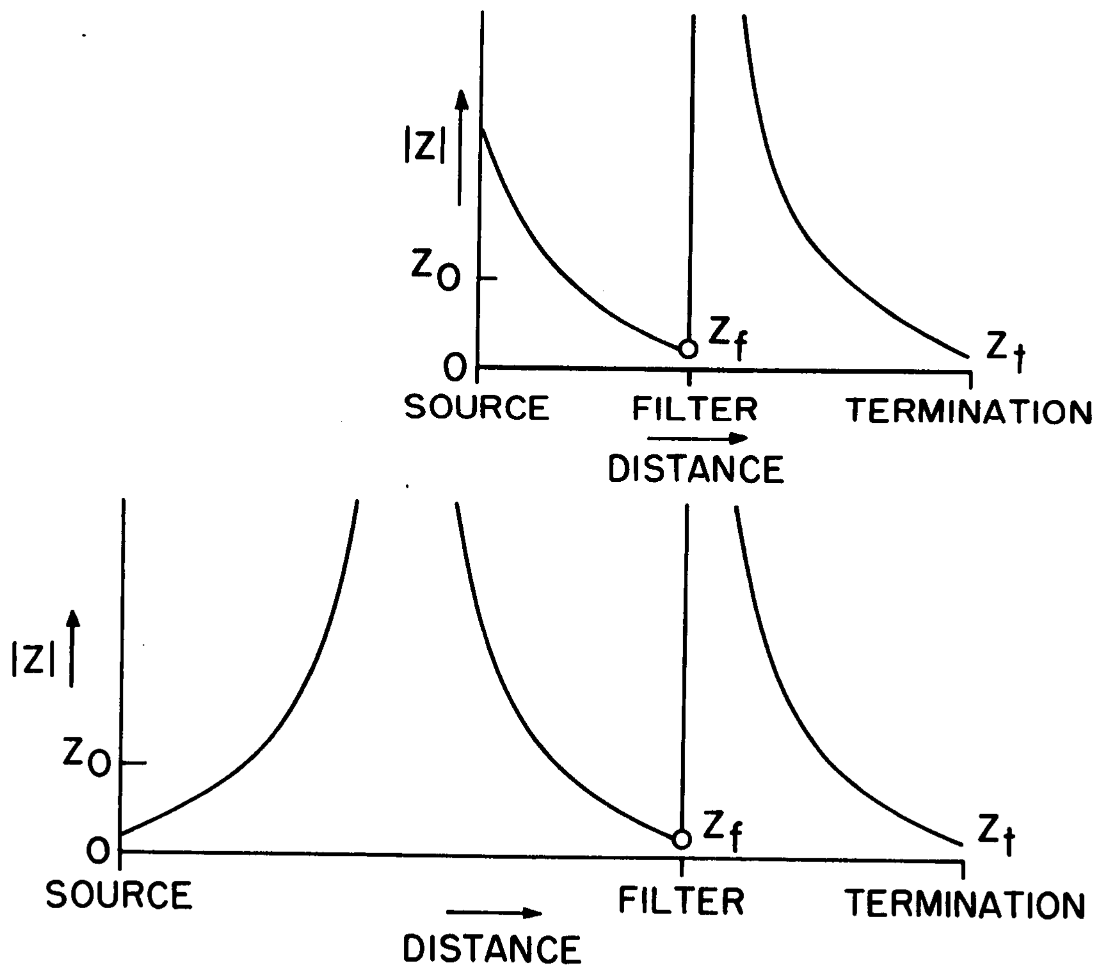
L-975-16 DAVIS 3/8/56

EFFECT OF TERMINATING IMPEDANCE



L-975-17 DAVIS 3-8-56

EFFECT OF SOURCE-FILTER LENGTH



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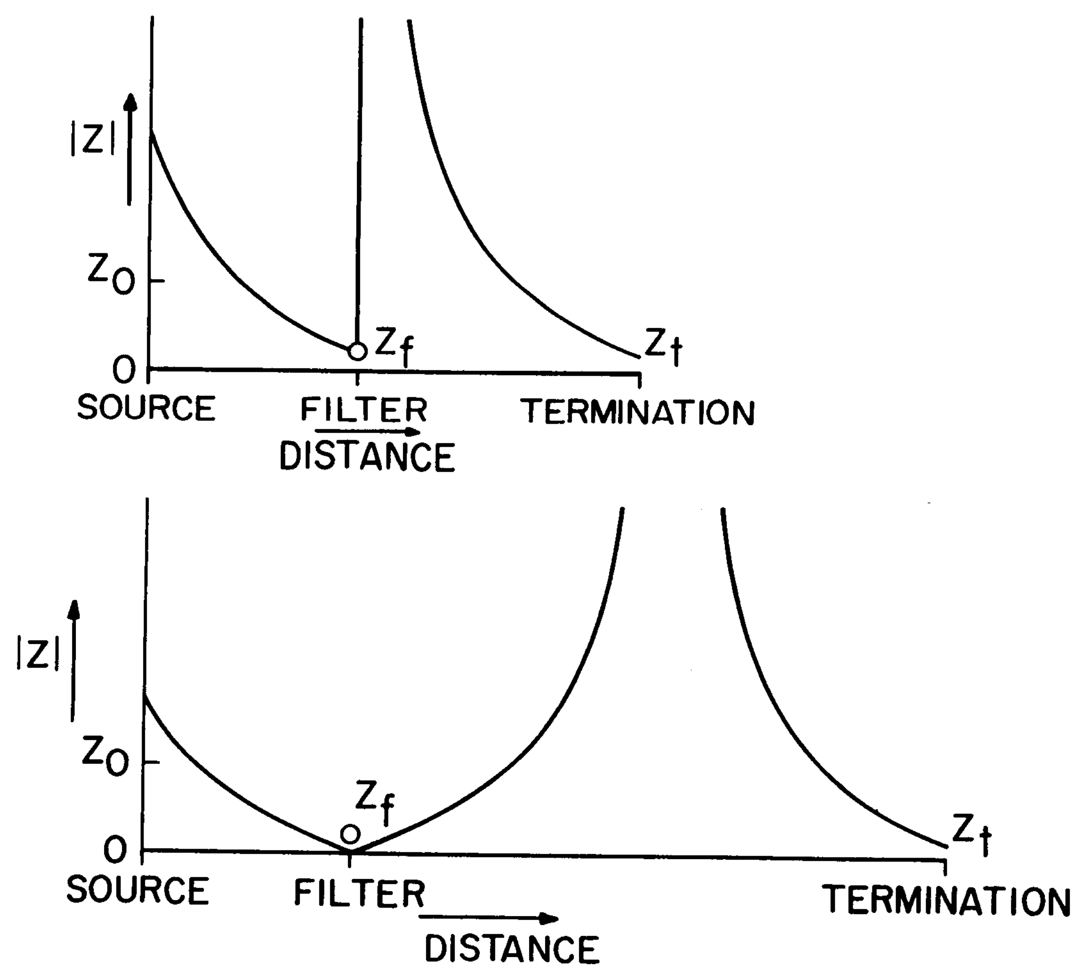
EFFECT OF FILTER-TERMINATION LENGTH

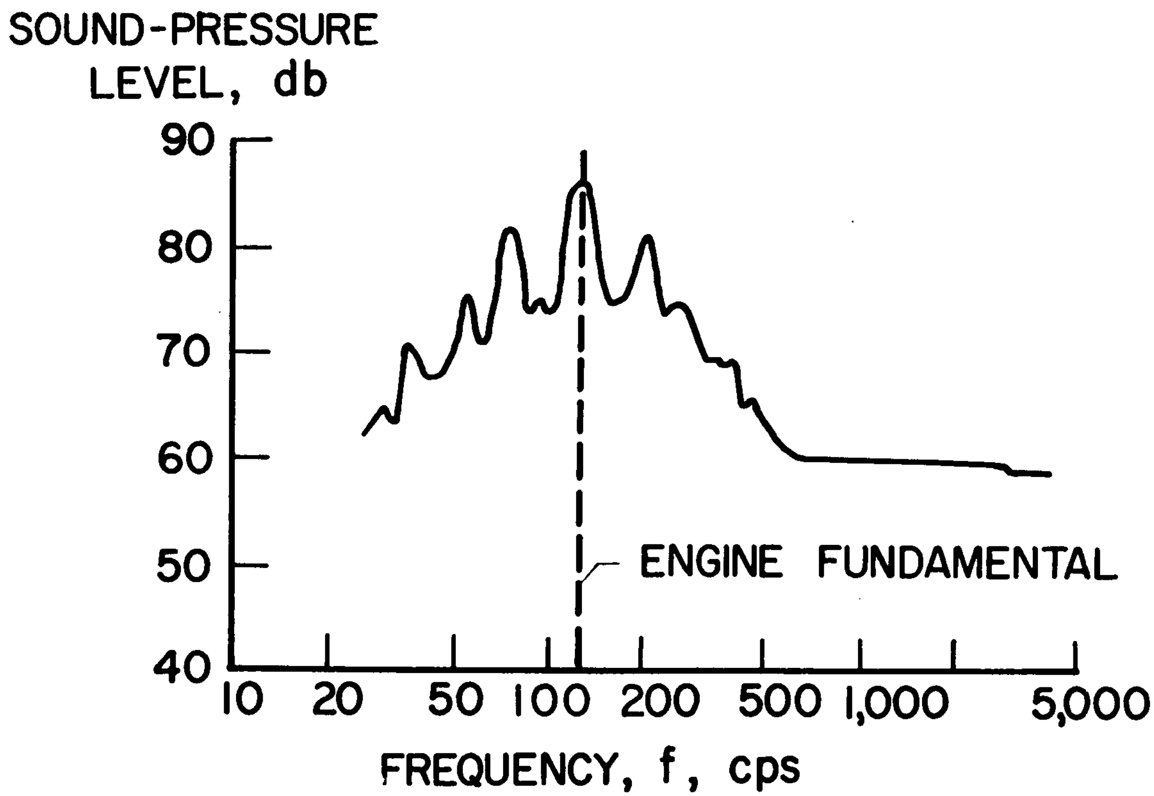


L-975-19 DAVIS 3/8/56

185 - HORSEPOWER HELICOPTER
(TAIL ROTOR REMOVED)

NACA L-63477.1 L-975-20 DAVIS 3/8/56

UNMUFFLED-HELICOPTER-NOISE FREQUENCY ANALYSIS



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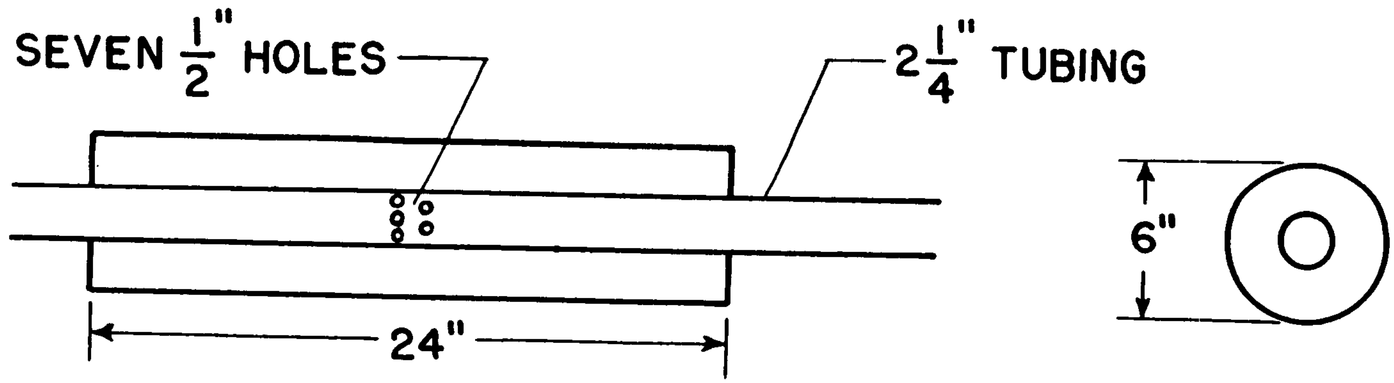
SKETCH OF HELICOPTER MUFFLER



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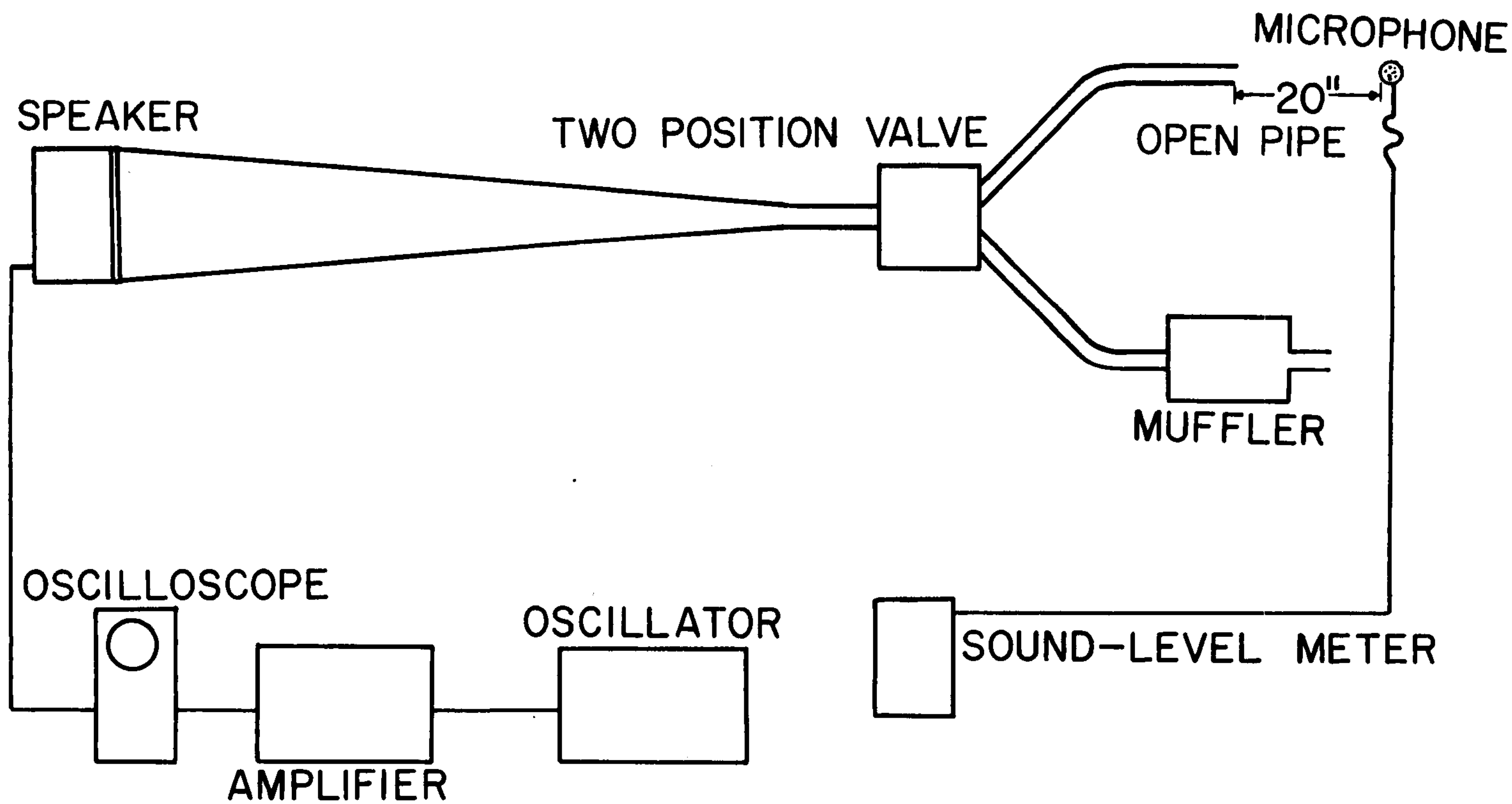
L-975-22

DAVIS 3/8/56

APPARATUS FOR COLD TEST OF MUFFLERS



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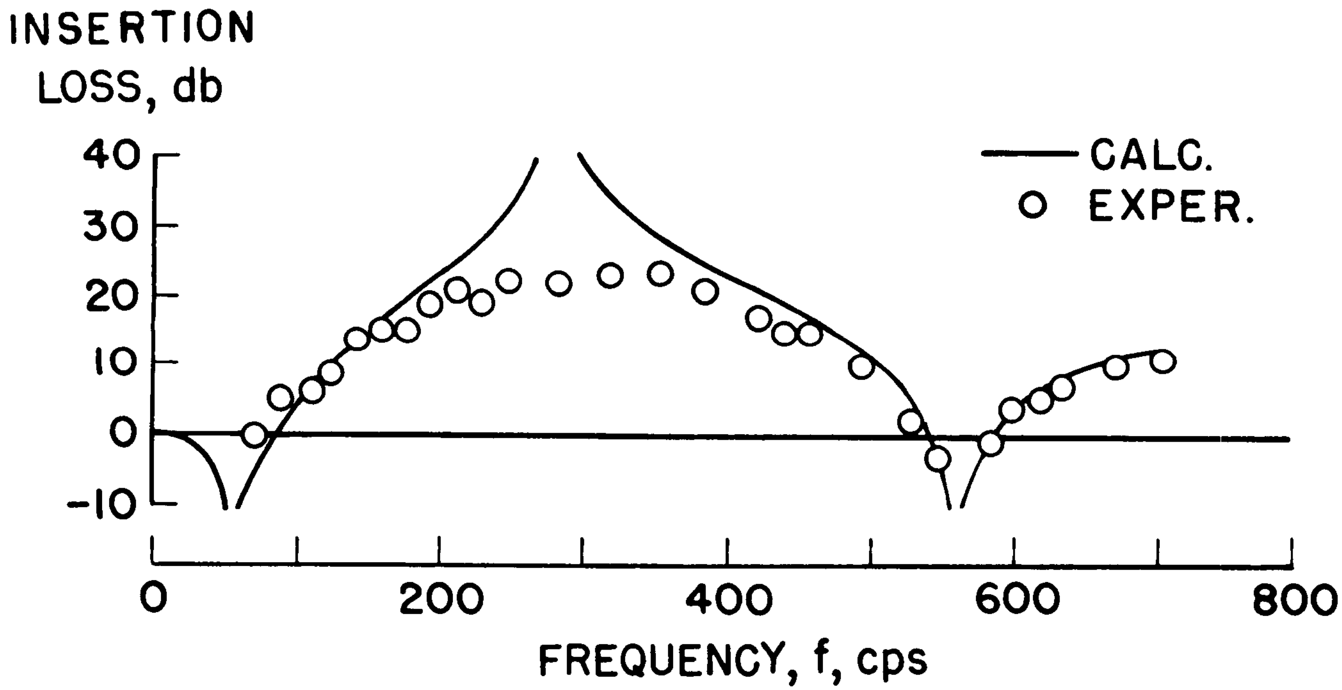
THEORY AND COLD-TEST DATA
HELICOPTER MUFFLER

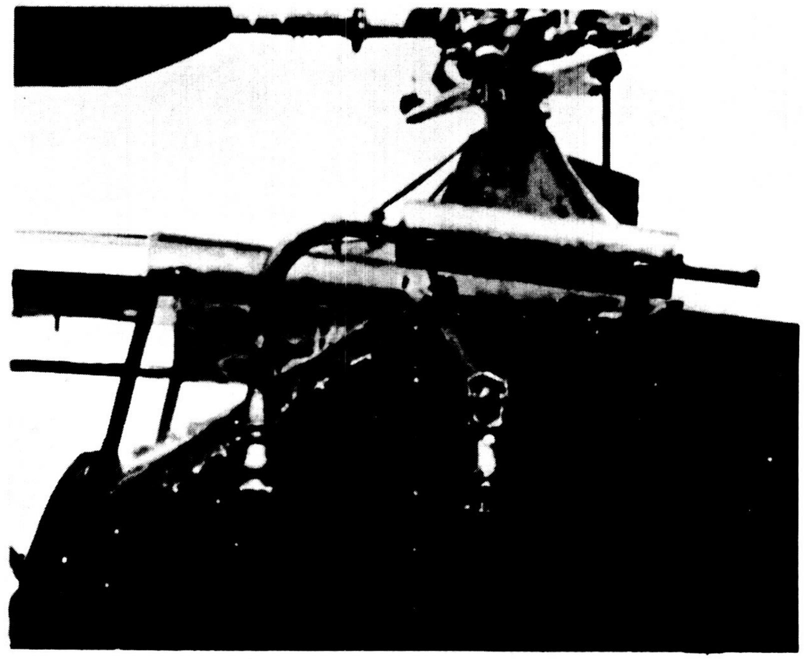


NACA

L-975-24

DAVIS 3/8/56

MUFFLER INSTALLATION



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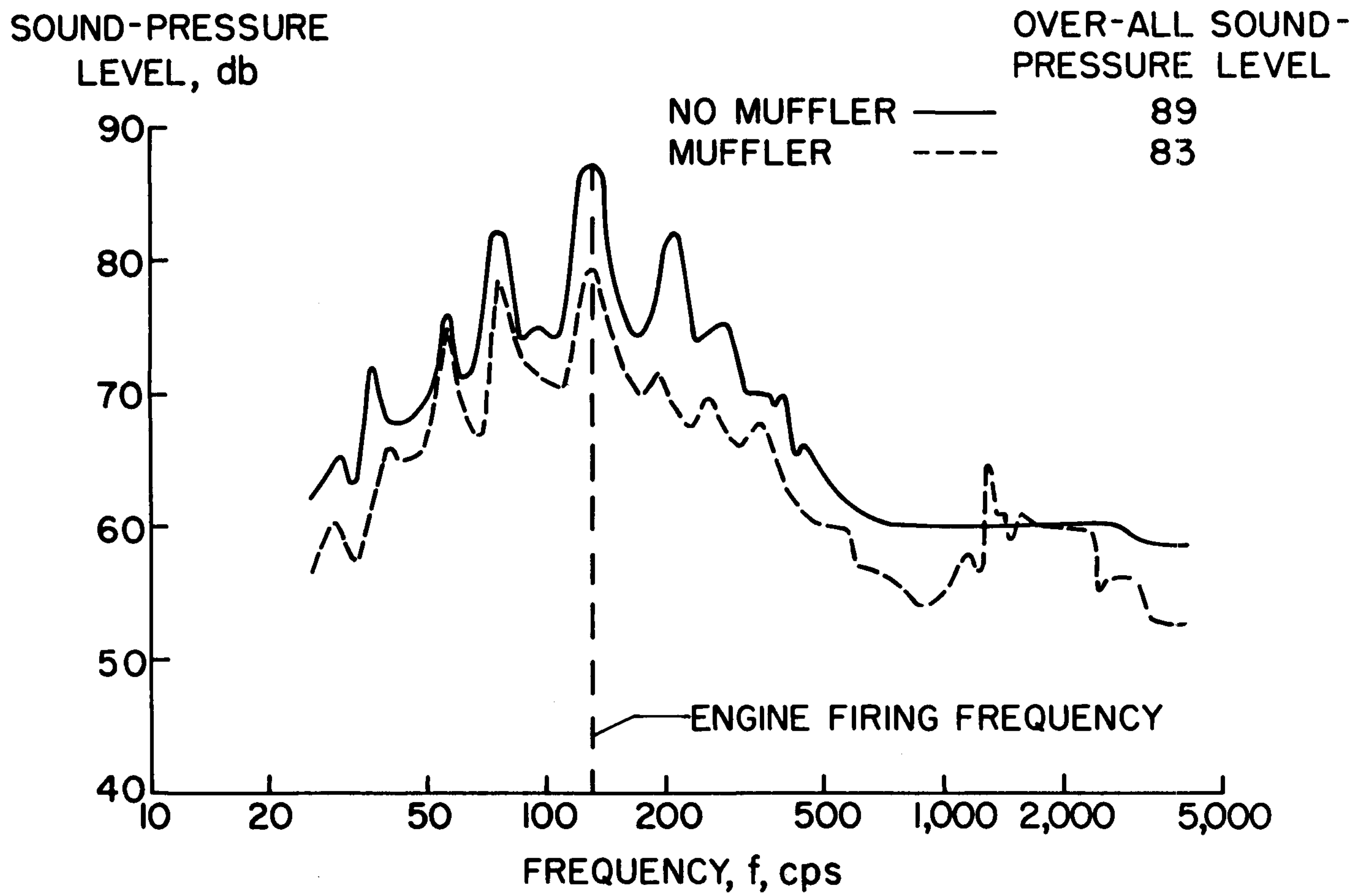
L - 975 - 25 DAVIS 3/8/56

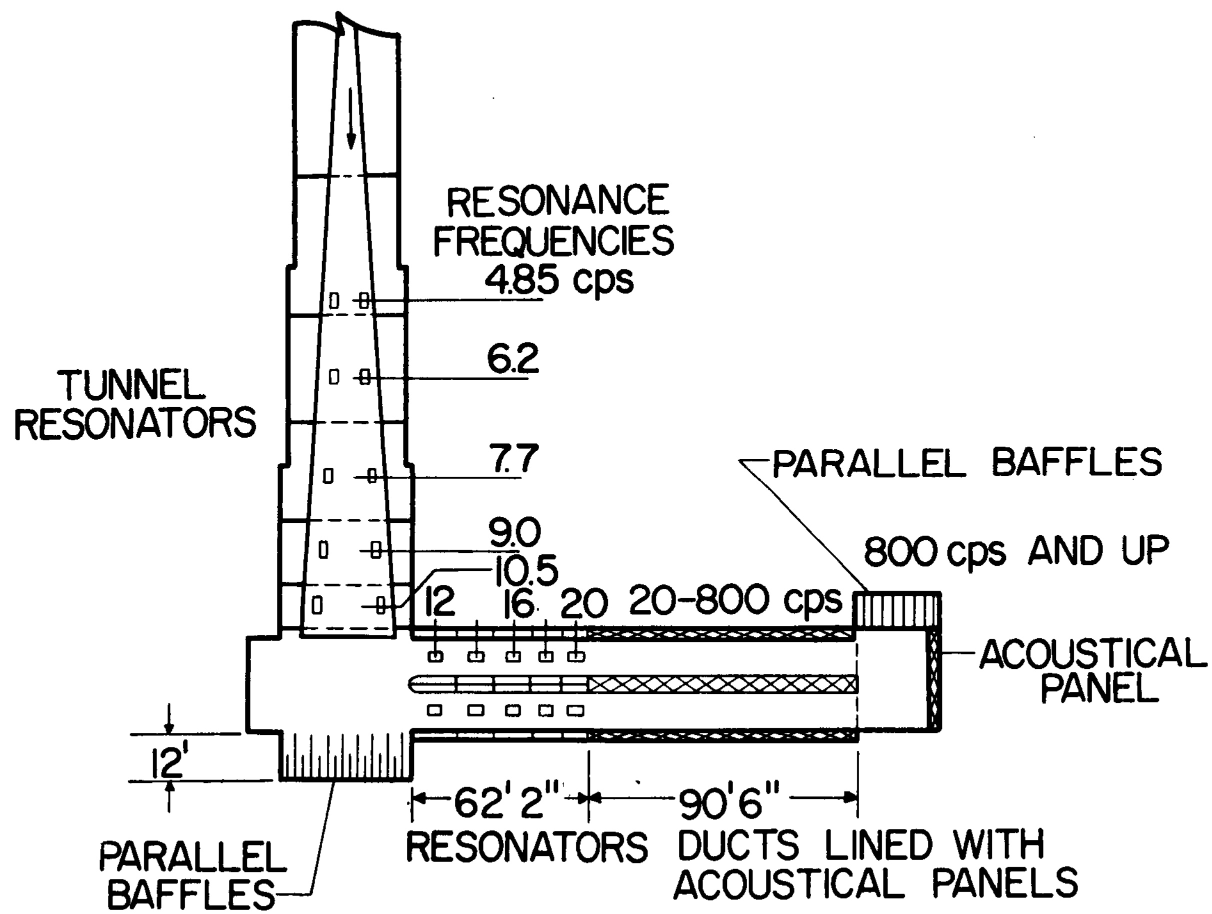
L-77948

EFFECT OF MUFFLER ON HELICOPTER NOISE



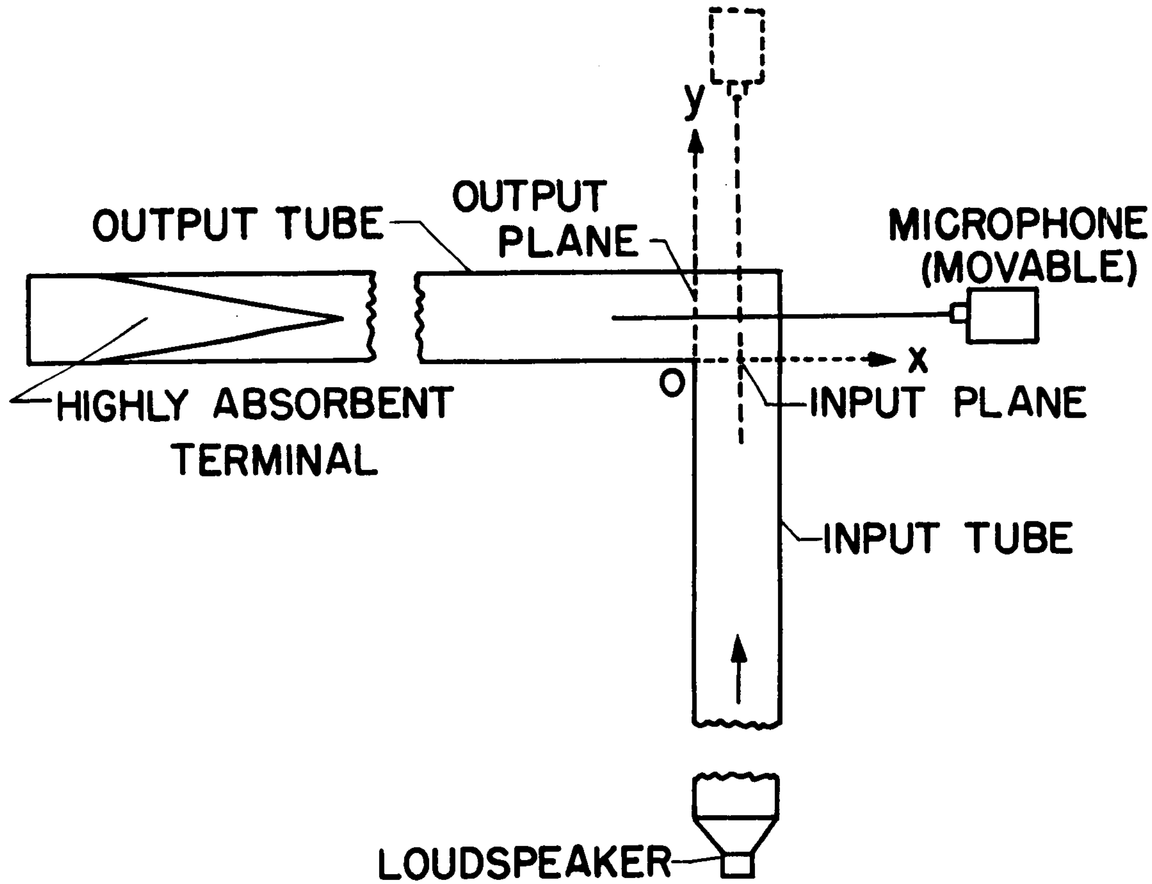
L-975-26 DAVIS 3/8/56

ACOUSTICAL TREATMENT FOR SUPERSONIC TUNNEL



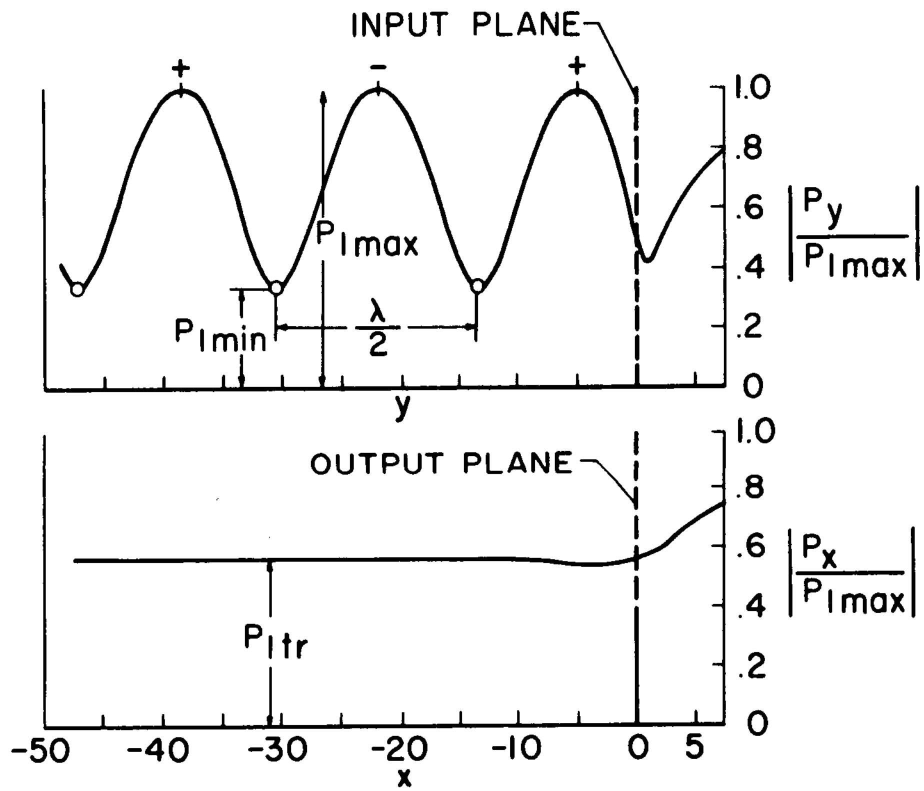
NACA L-975-27 DAVIS 3/8/56

APPARATUS FOR MEASURING TRANSMISSION AND REFLECTION FACTORS



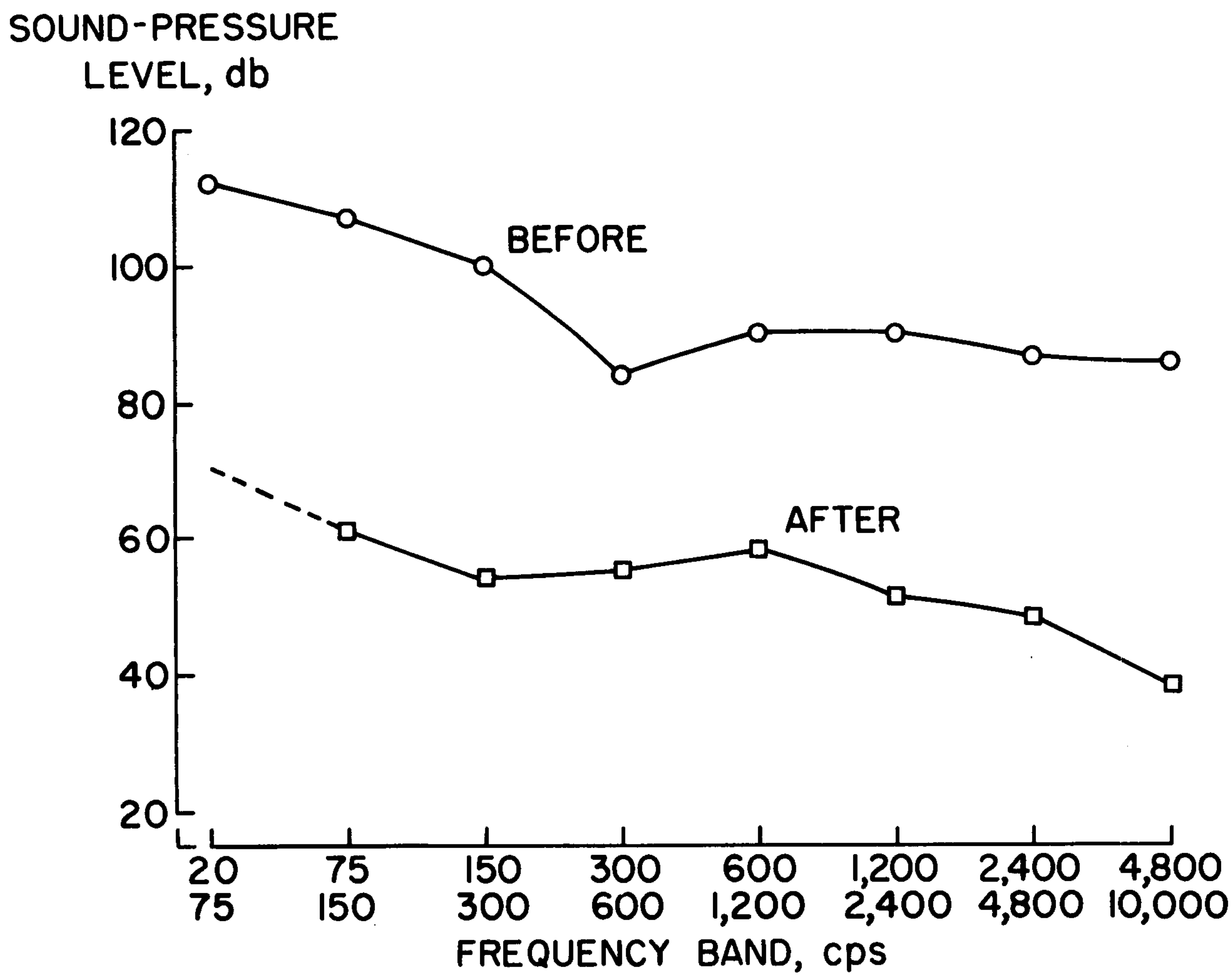
NACA L-957-28 DAVIS 3/8/5(

EXPERIMENTAL DATA AND DERIVED FACTORS



$$|R| = \frac{1 - \frac{P_{I\,min}}{P_{I\,max}}}{1 + \frac{P_{I\,min}}{P_{I\,max}}} \qquad |T| = \frac{2\,\frac{P_{I\,tr}}{P_{I\,max}}}{1 + \frac{P_{I\,min}}{P_{I\,max}}}$$

WIND-TUNNEL SOUND LEVELS



NACA L-975-30 DAVIS 3/8/56

SOME AUTOMOTIVE NOISE PROBLEMS

David C. Apps, Head
Noise and Vibration Laboratory
General Motors Proving Ground
Milford, Michigan

In discharging my onus to this group this afternoon, I think it would be helpful to all concerned if I started out with two or three organization charts to help you visualize the relationship of the Noise and Vibration Laboratory of the General Motors Proving Ground to the Proving Ground organization and to the echelon immediately above.

Figure 1 is an organization chart of the GM Engineering Staff directed by a General Motors vice-president, Mr. C. A. Chayne. In its broadest terms, the group on the left is responsible for advanced engineering developments of automobiles and automotive components. The group in the center of the Figure represents the special service groups such as accounting, machine shop facilities, etc. On the extreme right is a group providing services and facilities for the entire General Motors Corporation; these include Patent Section, Photographic Section, etc. in this group on the right is found the General Motors Proving Ground, and it is within the box entitled "Other Locations" since it is not located at the new General Motors Technical Center.

Figure 2 shows the Proving Ground organization directed by Mr. H. H. Barnes. By the standards of most of our manufacturing plants, it is a relatively small organization. It operates as a service facility for the entire Corporation, and as such there is no manufacturing done in this group. I might point out that included under the management of the General Motors Proving Ground at Milford is the Desert Proving Ground near Mesa, Arizona, and a Pikes Peak Engineering Test Headquarters at Manitou Springs, Colorado.

The Proving Ground organization is again divided into an Engineering Group and a Service and Facilities Group, with the Noise and Vibration Laboratory part of the Engineering Group.

Figure 3 shows the organization chart of the Noise and Vibration Laboratory. The dotted boxes indicate separate groups not yet created into separate entities. They are 1) an instrument development and calibration group on the left, and 2) an advance development group on the right. Each will ultimately have their own group head but the work is now handled as required by personnel of the other groups shown in solid boxes. We will dwell largely this afternoon on the activities and experiences of the three groups in the center indicated as performing these functions:

1. Noise tests, materials tests, and audiometry.
2. Noise problems.
3. Vibration and stress problems.

The group on the left handles all of our binaural record-playback testing, of which more later. They conduct tests to determent the vibration damping properties of automotive materials by the Geiger thick plate test. They conduct sound absorption tests by the standing wave closed tube method, and as a Proving Ground service they conduct audiometry tests on all Proving Ground personnel.

The center group, which is charged with solving noise problems, handles a wide variety of problems ranging from a ticking speedometer to the noise reduction program on the GM Aerotrain. The group on the right handles a wide variety of vibration and stress problems on the automobile as a whole and on various components or subassemblies. Both the SR-4 strain gage and brittle lacquer Stresscoat are used in experimental stress analyses.

I would like to make it clear that the Proving Ground Noise and Vibration Laboratory acts as a service, or a consultant group to the entire General Motors Corporation on noise, vibration, and stress problems. Actually, this is not quite as comprehensive an assignment as it appears inasmuch as we are not charged with the primary responsibility for the quietness of all GM products. This is to say, that our organization and facilities are available upon call when some responsible Divisional authority decides that we are to be consulted. Many of our Manufacturing Divisions have sound testing facilities and personnel of their own but use the more complete facilities and more experienced personnel of the Noise and Vibration Laboratory as the occasion requires. Our findings and recommendations on a given job are reported in a confidential report to the requesting Division or agency and in a strict sense are only recommendations. The final decision whether and how the recommendations are to be executed or put into production is the decision of the autonomous manufacturing Division.

It might be well to point out at this time that the automobile testing staff of the Proving Ground is a fact-finding body charged with determining automobile performance, economy, handling, braking and other characteristics of a cross section of American-made and some foreign-made automobiles. The findings of the Proving Ground regarding test results of both General Motors and competitive cars are made known by means of engineering reports. In no sense does the Proving Ground put a stamp of approval on a pre-production GM car indicating that it has passed a series of tests which make it suitable for production.

Let us start detailing some of our activities and some of our noise and vibration problem experiences. Turning to the first group on the left in the organization chart which I showed you, the group which conducts noise tests and materials tests, let us start out by saying that we do a considerable amount of noise evaluation tests for the control of noise in the automobile. We do this almost entirely by a binaural record and playback procedure which our group at the Proving Ground was largely responsible for promoting. We started using a record-playback procedure in about 1940 by using an instantaneous acetate disk recording method. In 1947, when wire recorders of the instrument class became available, we were glad to rid ourselves of the incubus of mechanical problems attendant upon the use of a disk recorder in a moving car or truck.

In 1951, we induced a tape recorder manufacturer to provide us, on a loan basis, a two-channel recorder, which we understand had been made for the Navy for some unrevealed or confidential use. The improved realism that we obtained with the binaural system, especially for inside-car recordings, convinced us of the potentialities of this system. We now have over four years of experience and have evolved a sensitive noise evaluation technique. We feel that we can sense, or detect, smaller differences than is possible by any other known method, and, furthermore, immediately and directly evaluate the importance of these changes in terms of customer reaction.

Figure 4 shows a view of a car under test and a station wagon equipped with a binaural recorder. The binaural microphones are placed in the test, or lead car; the microphone signals are sent to the recording mechanism through the cable and cable payout device on the roof of the station wagon.

Figure 5 shows an interior view of the recording mechanism set up in the station wagon, and Figure 6 shows a jury listening to a series of recordings to evaluate them. The jury listens to these recordings in an ABA sequence and indicate their choice, or preference, by an electrical indicator which insures independent judgments. A paper covering the use of a binaural recorder has been published in the Journal of the Acoustical Society of America (Vol. 24, p 660; Nov., 1952) and I might refer those interested in some of our techniques to that article.

Figure 7 shows a method used to determine the relative importance of the front and rear suspension in transmitting road noise or rumble. The Opel, second car in the train, is under test, having its rear wheels suspended from the pavement by the tractor following it. The lead car is towing the test car and tractor. The recording station wagon at the rear moves along under its own power. The road rumble generated by front wheels alone is compared with that generated by the rear wheels and by all four wheels. These tests indicate on which end of the car to emphasize suspension isolation.

The recording and playback procedure discussed above is a routine procedure. The methods of controlling automotive noises of various kinds is anything but routine. The group who are responsible for conducting these tests find themselves compiling a considerable backlog of experience which they have gained over the years on a variety of problems on a wide range of automobiles, trucks and buses. You will recognize this procedure as a noise control procedure rather than a noise reduction or elimination procedure based upon reduction of noise at the source.

I might add, in passing, that this group contributes in an important way to the acceptance of GM cars, and their recommendations may easily involve as much as $100,000 to $1,000,000 per model year on a single make of car.

We might summarize the principles relating to the control of automotive engine noise by saying that engine noise as heard by the passenger is controlled by the use of an underhood absorption blanket, but the use of a dash mat or pad on the firewall, together with a floor mat or carpet which provides damping as well as increased transmission loss. In this connection, it is interesting that the usual jute floor mat which frequently has an impervious rubber padding on top provides more damping if it is laid directly on the steel floor than if it is cemented thereto. A slight reduction of engine noise is also obtained in some car models by providing an acoustic blanket behind the headlining in the passenger compartment. In addition, this acoustic blanket provides a sense of luxury, or comfort, much like heavy carpeting and drapes do in an ordinary room.

In the automotive product, it is important to provide for the control of impact noise as elicited by rain or hail striking the roof or by the prospective customer thumping or knuckling the doors and roof when the car is in the sales room. We have found that the quality sound (or lack of tinniness) excited by local impact is due entirely to the surface density of the damping material applied rather than to its damping qualities. This statement is true on the sheet metal thicknesses used on automotive bodies. With thicker sheet metal gages, the surface density of the deadener may be less important and the damping properties may begin to assume relatively more importance.

The materials testing function of this same group includes determination of sound absorption coefficients by the classical closed-tube standing wave method. At the present time, we do not have a reverberation room available for making absorption checks. This group also determines vibration damping properties of various automotive materials such as asphalt saturated felts, spray-on materials of the cut-back or water-soluble varieties, etc.

Figure 8 shows the Geiger thick plate apparatus on which this is accomplished. Very briefly, all materials are tested on a $\frac{1}{4}$-inch-thick steel plate, 20 inches square. This plate has a resonant frequency of approximately 150 cycles, which is excited by a magnetic drive coil beneath the plate. To evaluate the vibration damping properties of a material, the plate is excited at resonance, the drive removed from the drive coil, and the decay of the vibration of the plate is recorded on a magnetic oscillograph or evaluated by an automatic device. The result is expressed in terms of decibels of decay per second. Needless to say, this is not an absolute method but rather it provides comparative data. We do not find it necessary to determine viscous or elastic moduli, which are perhaps more important to the people who are responsible for developing such damping materials and compounds.

Figure 9 shows a view of an experimental setup for determining the transmission loss of a Scenicruiser rear window. The problem here was to determine the relative transmission loss of two $\frac{1}{4}$-inch plate glass rear windows with 3/8-inch air space, compared with a single $\frac{1}{2}$-inch glass window. We thought we could predict the answer on this one without even running the test. It was our feeling that the two leafs of glass would provide more transmission loss than the single glass because it would have the same overall surface density and, in addition, would have the compliance offered by the 3/8-inch intervening air space. Actually, our test using white noise as a source showed that the overall transmission of the single $\frac{1}{2}$-inch glass was somewhat better than the two individual leafs. In more detail, it was somewhat poorer at low frequencies but better at high frequencies.

Figure 10 shows the rear end view of a GMC transit-type coach in which the combustion noise of the diesel engine is controlled by essentially sealing up the engine compartment and adding about 30 square feet of sound absorptive material held in place by perforated sheet metal. This, of course, is an obvious and time-tested method of noise control and presents no particular problem except that of permitting the coolant air to escape while retaining the noise. This was done by using a rather primitive type of lined acoustical duct which took the form of a mushroom with an absorbent lower surface placed immediately above and overlapping a hole in the engine compartment baffle pans.

One of the more interesting jobs that we have worked on lately is that of noise control on the new GM Aerotrain. As you have no doubt read in the press, two of these 10-car lightweight trains have been made and have been leased to the railroads for their trials. They intend to place them in passenger revenue service in order to evaluate the concept of a lightweight, high speed, low cost train. It may interest you to know that these cars, which carry 40 passengers each, weigh about 1,000 pounds per passenger seat as compared with about 2,000 pounds per passenger seat in a standard type railway coach. The cost of the train is estimated to be about one-half the conventional train cost including locomotive. It is estimated that the maintenance cost will be about one-half, and this comes about largely because the superstructure of the coaches is cheap enough so that a new superstructure, or body, can be installed for less than the conventional $70,000 per coach refurbishing cost. The fuel costs are also about one-half the fuel costs of the average train. Because the train weighs less, it obviously takes less power to haul it. The 1200 horsepower locomotive will pull the train at 100 mph.

At the request of our Electromotive Division, we made a study of the inside noise of these coaches. At the time we were called in, all 20 of the cars had been built and, in fact, were due to be shipped out to the railroads about two weeks later. We made magnetic tape recordings of the noise at various locations of the car, both inside and outside, at different speeds and with different track conditions. We analyzed these tapes and conducted further tests with the cars stationary at the plant. Figure 11 shows one of these plant tests in which a battery of 7 speakers on plywood baffles was placed in a pit beneath the car. A short loop of tape taken from one of the outside noise recordings made on the Rock Island tracks was then played through the speakers to simulate the outside noise. Our objective here, of course, was to reproduce the sound without any vibration due to the moving train, or in terms more understandable to this group we want to evaluate the airborne versus the structureborne noise.

On the basis of these studies, we were able to devise changes in the structure and in the design of some of the running gear components on an experimental basis. The car was operated on the test track at the plant and the experimental changes were evaluated for effectiveness in controlling the noise. While I am not at liberty to indicate the nature of the changes made, I can say that we made a sizeable reduction in the noise level inside the car. Because of the time limitation, it was not possible to incorporate these changes on the cars before they left the plant for their demonstration runs and subsequent release to the railroads. These changes have since been made on both trains.

The above are examples of noise control by preventing its transmission or by absorbing or damping it somewhere along the path between its source and the observer's ears without making any attempt to improve the mechanism or device to reduce the noise at the source.

On the basis of our experience, it isn't easy to pull from our files a half dozen examples where noise was reduced by a redesign of the mechanism, thus controlling it at the source. I cite a few examples below, two of which we spoke of in some detail at the M.I.T. Summer Symposium on Noise Reduction last August. Therefore, this coverage will be extremely brief.

One example concerned the control of power steering noise, in which it was found that the rather soft yielding rubber hose connecting the power steering pump to the steering unit was the source of trouble. When such a soft-walled tube yields in a radial direction due to the alternating fluid pressure, the length is changed at the same frequency, thus providing a source of mechanical excitation if the hose is drawn tight. The cure is to put a bend in the hose so that the alternating changes in length can be harmlessly dissipated.

The other example concerned noise reduction at the source in the small electric motors which drive the windows up and down in our present-day cars. It was found that winding short-circuited coils having the same pitch as the commutator winding accomplishes a sizeable noise reduction. The number of turns of wire and the size of the wire to obtain (what was probably close to) a maximum reduction was determined experimentally. Incidentally, this is a case where the experimental method was used without benefit of analytical treatment whatsoever.

Turning now to the activities of the group responsible for studying vibrations and making stress analyses, literally hundreds of such jobs have been accomplished during the past few years, and we will briefly run through a few which are more or less typical.

Figure 12 shows a rear leaf spring of an automobile disassembled with strain gages applied, ready for assembly. Figure 13 shows the strain gage amplifiers and multichannel magnetic oscillograph used for recording the strains and other dynamic quantities. Figure 14 shows the instrumentation devised for the measurement of shock absorber behavior. The quantities measured in this instance were the vertical spring force, the spring displacement, the shock absorber damping forces, the shock absorber piston velocity, shock absorber piston displacement, and two hydraulic pressures, one each in the compression and rebound chambers. Tests such as these were discussed by H. W. Larsen of our Laboratory before the SESA in Milwaukee, Wisconsin, in May, 1953, and published in Vol. 13, No. 1, of the Proceedings of the SESA.

Figure 15 shows a crankshaft and flywheel in which it was desired to measure stresses. Inasmuch as the automatic transmission behind the flywheel prevented attachment of sliprings rearward, the strain gage lead wires had to be threaded through holes in each main bearing journal and crankshaft throw journal to finally emerge at the front end of the crankshaft, where there was room available for mounting the small slipring assembly shown. This Figure is shown to demonstrate how important ingenuity and persistence are in devising experimental setups to solve certain problems, a fact with which you gentlemen I am sure are well acquainted.

Figure 16 shows a gyroscope mounted on the front axle of a truck to determine the axle windup due to brake application.

Figure 17 shows an example of the use of Stresscoat on a front solid axle to determine the bending stresses due to brake application. In this connection, I might say that in reading the literature we get the impression that the use of Stresscoat has been largely limited to laboratory use where temperature and humidity can be controlled. However, we have taken Stresscoat out of the laboratory by sheer necessity in order to locate areas of maximum stress as well as the orientation of the stress. We find that we have to apply it to frames, spring hangers, axles, suspension parts, sheet metal and bodies, etc., and gladly yield somewhat on the accuracy in order to take advantage of the merits that this coating provides. I might say, too, that we do a considerable amount of dynamic testing with Stresscoat immediately following which we examine the coating for cracks and mark them and photograph them as shown in this Figure.

We have found an electrodynamic shaker to be an almost indispensable aid in vibration and stress problems.

Figure 18 shows a small 10-pound shaker applied to a starter to determine the resonant frequency and the mode of vibration. The shaker study followed from an investigation of a noise period in which the starter vibration was induced by the motion of the engine.

Figure 19 shows the use of a 50-lb shaker in a fatigue study on an oil filter housing or can.

Figure 20 is proof positive that the noise and vibration business need never be a narrow and drab one. This road is intended to be equivalent as far as noise and vibration are concerned, to certain basalt block roads found in Germany. Our German subsidiary, Adam Opel, found themselves occasionally in trouble due to a roar when traversing these roads at certain speeds. We found it desirable to duplicate those roads in this country so that prototype cars could be checked out for this quality. Plaster casts of several typical sections of the German roads were made by the subsidiary and sent to us. We in turn proceeded to make a plaster cast of the contours of 8 or 10 of these typical blocks from which gray iron castings were made to use for the headers in a regular cement block machine. These 8 block patterns were then laid in a random fashion in two strips with concrete foundations and curbs in order to prevent settling and to maintain stability.

The problem in specifying the nature of the block elements was to obtain a frequency excitation which was not too sharply peaked with speed, such as would follow if all of the road elements had the same dimension in the direction of travel of the car. By randomly spacing the 8 differently sized and shaped blocks, we were able to obtain a peaking quality quite similar to the German roads. That this was attained was determined by a comparison of magnetic tape recordings made on the German roads with similar tape recordings made on the domestic counterpart.

In the automotive industry, we find that in spite of the vast array of electronic instrumentation available commercially, we still have certain kinds of noises which are not properly evaluated by commercial instruments. One of these that I might mention is road rumble, or road noise. Although we have made a start to devise a road noise meter, our development is not completed. When we wish to compare the relative quality of, say, 10 or 20 cars as we do each year at the Proving Ground, we still depend upon a jury, or panel, to ride these cars and evaluate them on an arbitrary scale.

One disturbance which we have taken out of the unmeasurable category is audible tire thump. This is actually due to the interference, or beating, of two adjacent harmonics of wheel rotational speed. To measure this disturbance, we devised an instrument called the Tire Thump Meter, shown in Figure 21, which basically measures the scallop or the depth of the modulation, if you wish, of the resulting beat envelope after passing through a bandpass filter of 20 to 70 cps. This Tire Thump Meter was discussed at an SAE meeting in Detroit in March, 1955, and it was subsequently published in the SAE Transactions, Vol. 63, page 787 (1955).

These are the problems of the automotive noise and vibration engineer which present a continuous and interesting challenge. We feel akin to you in this group who meet a similar flow of noise problems, with perhaps a slightly different twist here and there. We are as proud as you must be, to belong to that rather limited engineering fraternity dedicated to the relief of man from that raucous reminder of a mechanized society.

GENERAL MOTORS CORPORATION • • • ENGINEERING STAFF
ORGANIZATION CHART

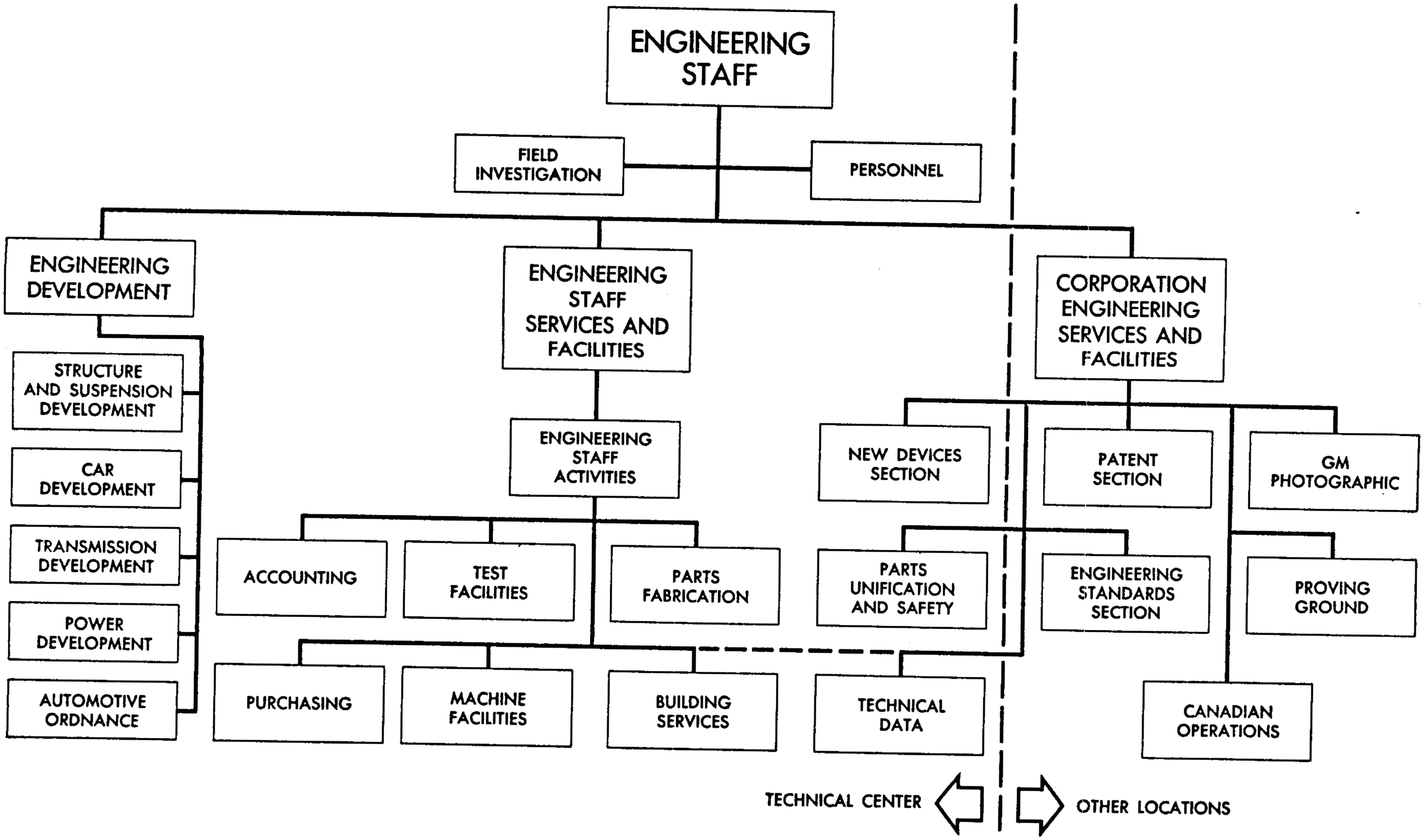
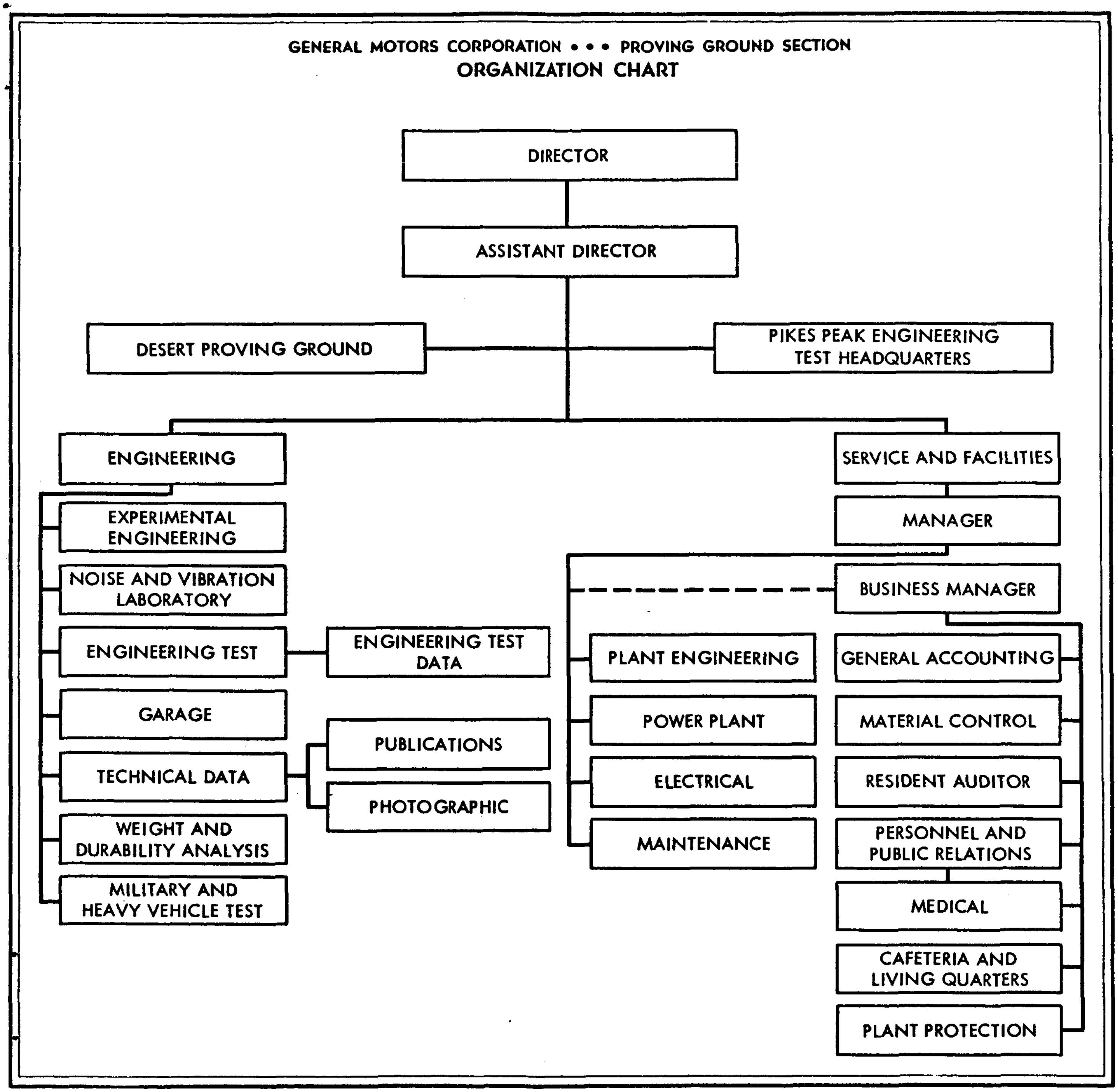


FIGURE 1


GENERAL MOTORS CORPORATION • • • PROVING GROUND SECTION
ORGANIZATION CHART
DIRECTOR
ASSISTANT DIRECTOR
DESERT PROVING GROUND
PIKES PEAK ENGINEERING TEST HEADQUARTERS
ENGINEERING
EXPERIMENTAL ENGINEERING
NOISE AND VIBRATION LABORATORY
ENGINEERING TEST
ENGINEERING TEST DATA
GARAGE
PUBLICATIONS
TECHNICAL DATA
PHOTOGRAPHIC
WEIGHT AND DURABILITY ANALYSIS
MILITARY AND HEAVY VEHICLE TEST
SERVICE AND FACILITIES
MANAGER
BUSINESS MANAGER
PLANT ENGINEERING
POWER PLANT
ELECTRICAL
MAINTENANCE
GENERAL ACCOUNTING
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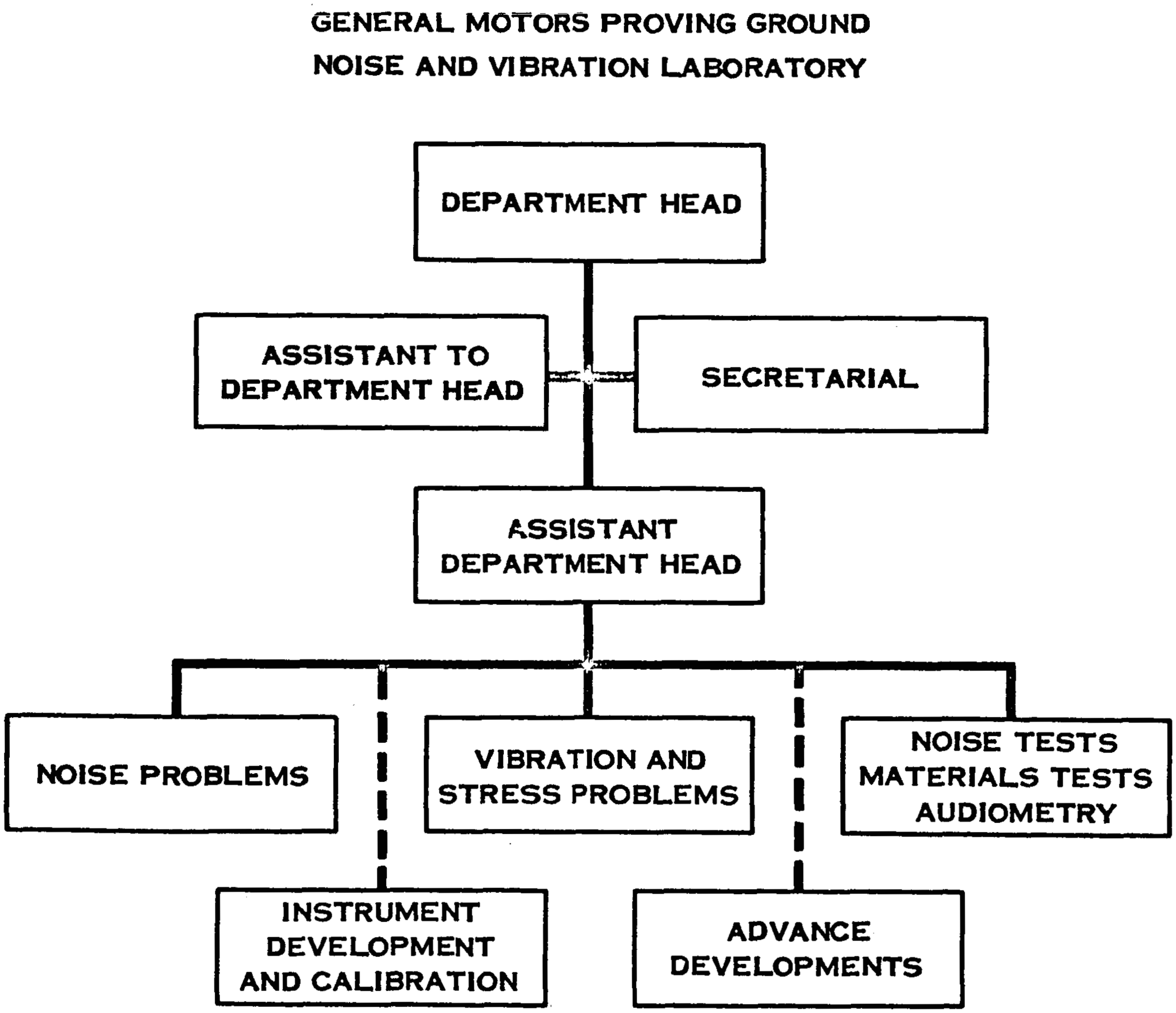
RESIDENT AUDITOR
PERSONNEL AND PUBLIC RELATIONS
MEDICAL
CAFETERIA AND LIVING QUARTERS
PLANT PROTECTION

FIGURE 2


GENERAL MOTORS PROVING GROUND
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DEPARTMENT HEAD
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NOISE PROBLEMS
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NOISE TESTS MATERIALS TESTS AUDIOMETRY
INSTRUMENT DEVELOPMENT AND CALIBRATION
ADVANCE DEVELOPMENTS

FIGURE 3

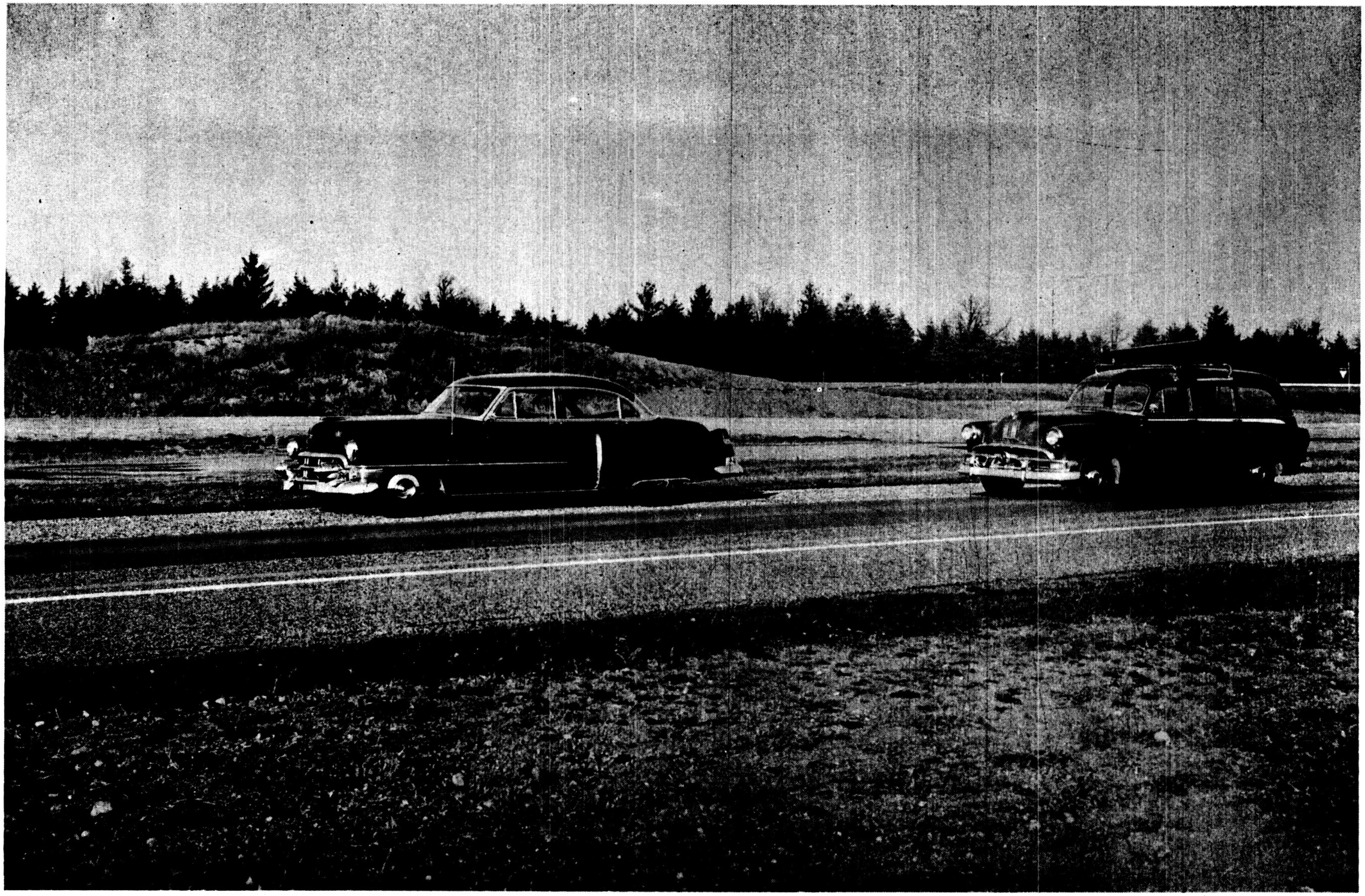

Figure 4 Binaural Noise Recording on Test Road. Lead Car is Under Test. Note Cable Payout on Station Wagon.

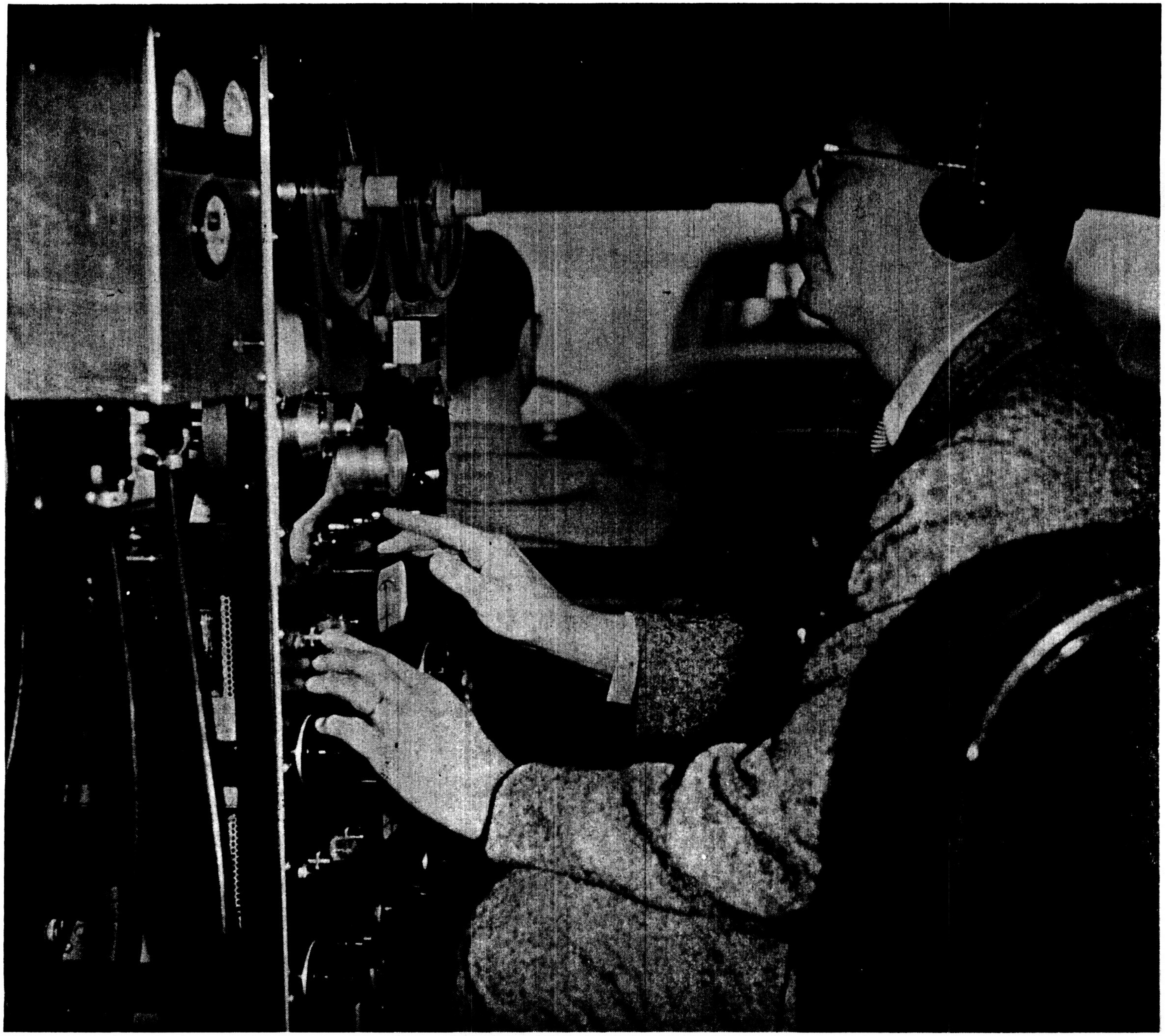

Figure 5 Binaural Tape Recorder in Station Wagon.

Figure 6 Jury, or Panel, Evaluating Binaural Recordings.

Figure 7 Four-Unit Train to Determine Relative Importance of Front and Rear Suspensions on Test Car, Second from Front.

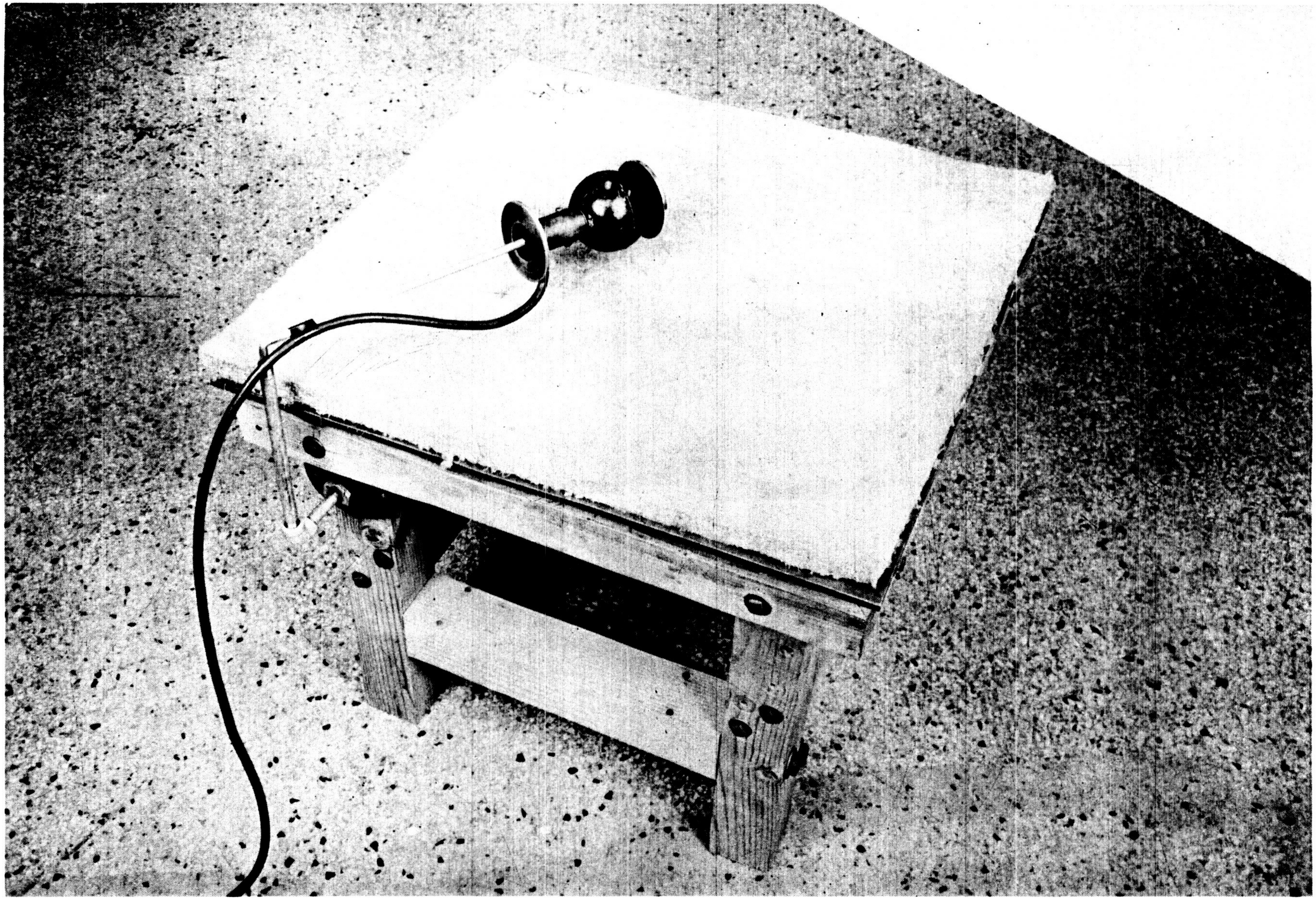

Figure 8 Geiger Thick Plate Vibration Damping. Microphone Senses Free Vibrations of Steel Test Plate, Shown with Test Material on Top. A Magnetic Drive (Not Visible) Excites the Plate at Resonance.

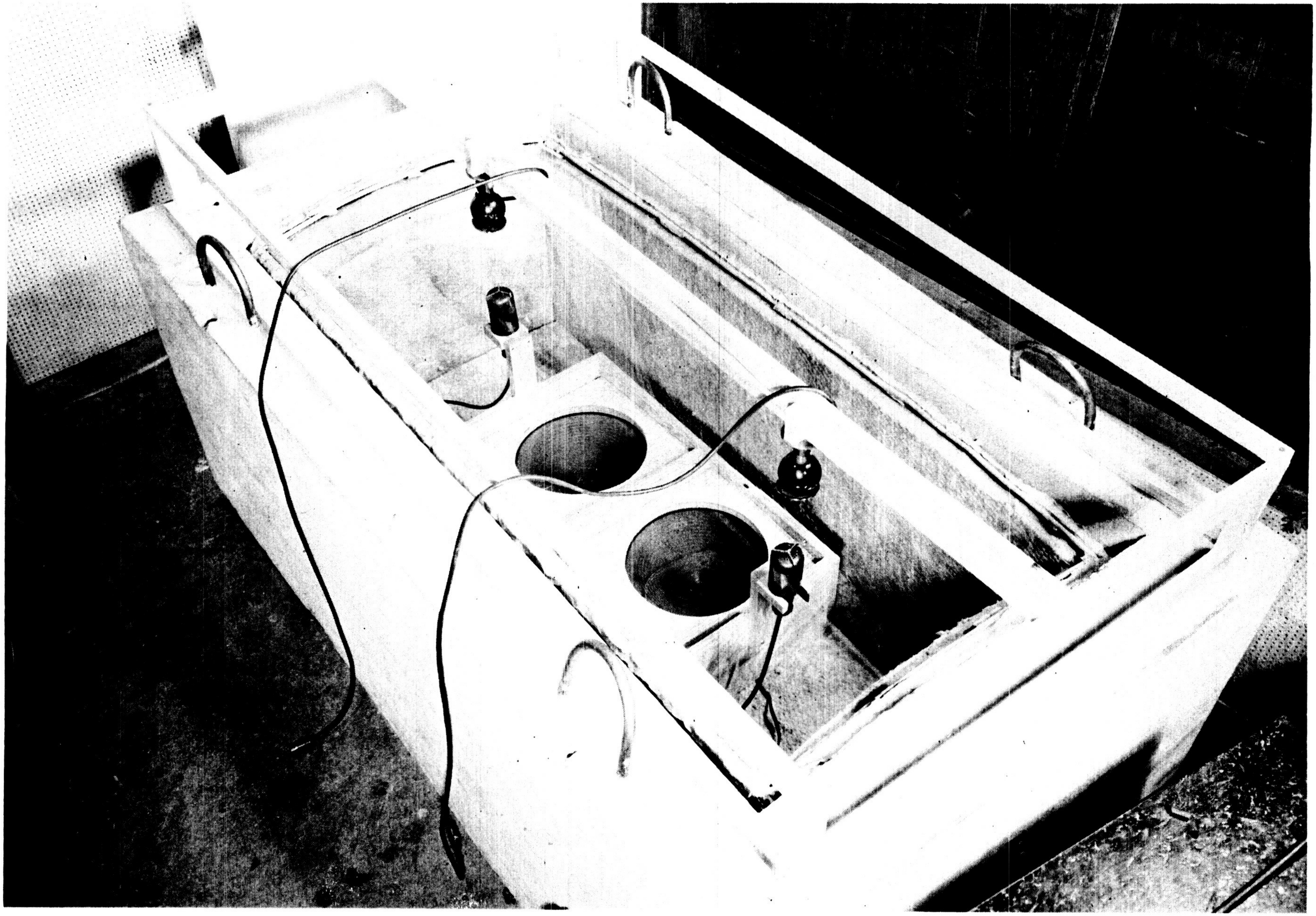

Figure 9 Concrete Box for Determining Transmission Loss of Rear Windows of Motor Coach Using "White Noise." Several "Outside" Microphone Readings are Averaged.

Figure 10 Engine Compartment of Transit-Type Diesel Coach, Showing Sound Absorption Lining and Retainers.

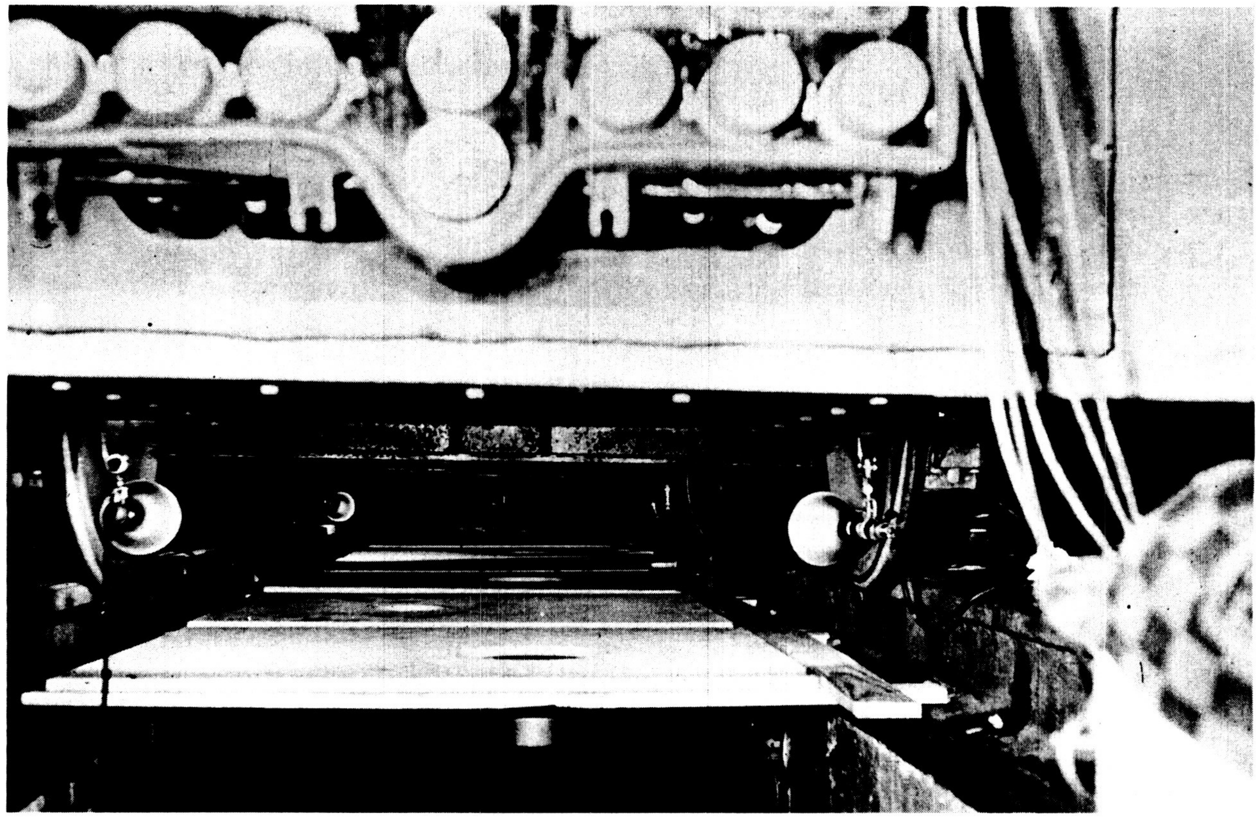

Figure 11 View Underneath GM Aerotrain Coach Showing Battery of Speakers on Plywood Baffles Used for Stationary Simulation of Moving Train Noise.

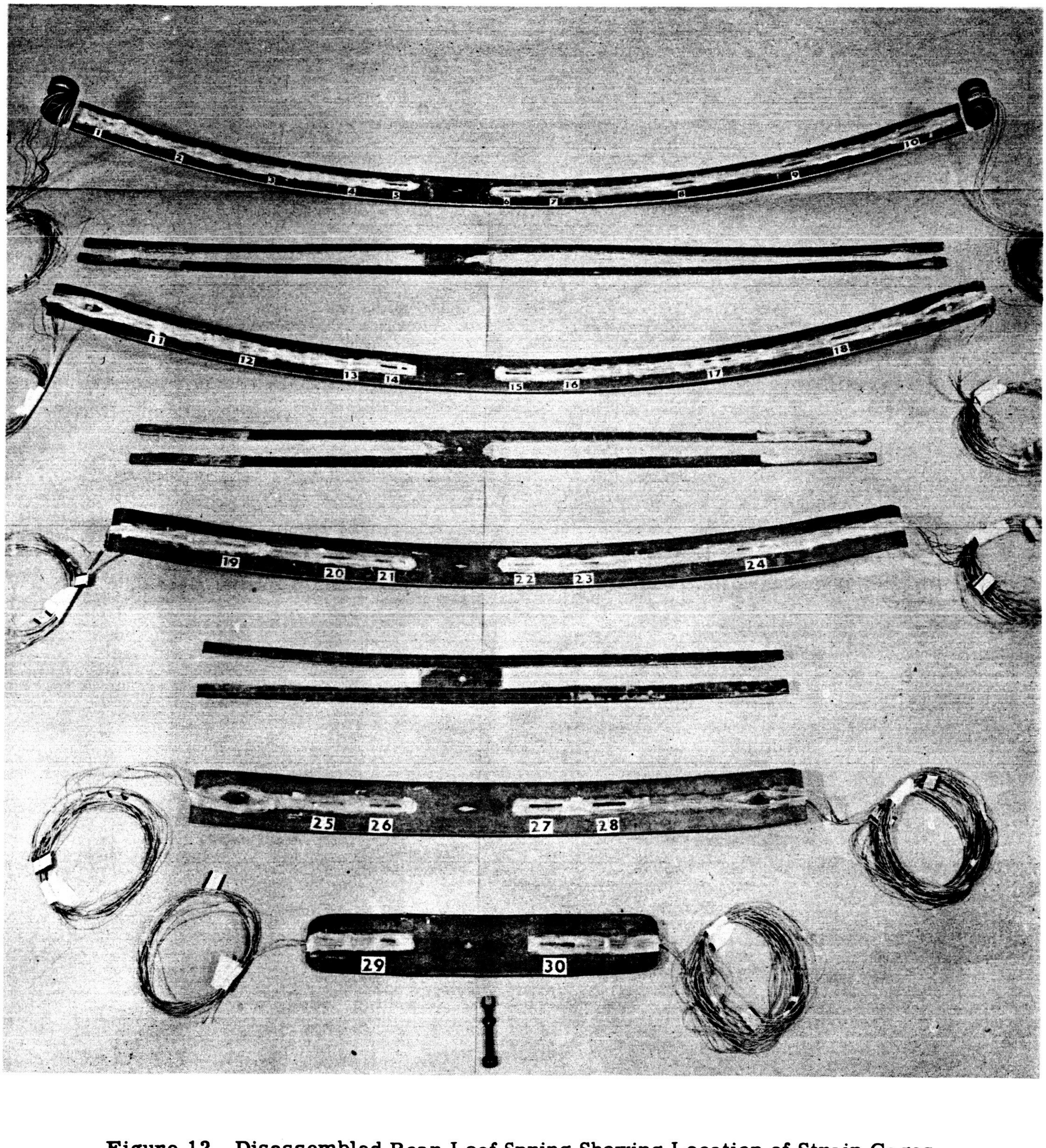

Figure 12 Disassembled Rear Leaf Spring Showing Location of Strain Gages.

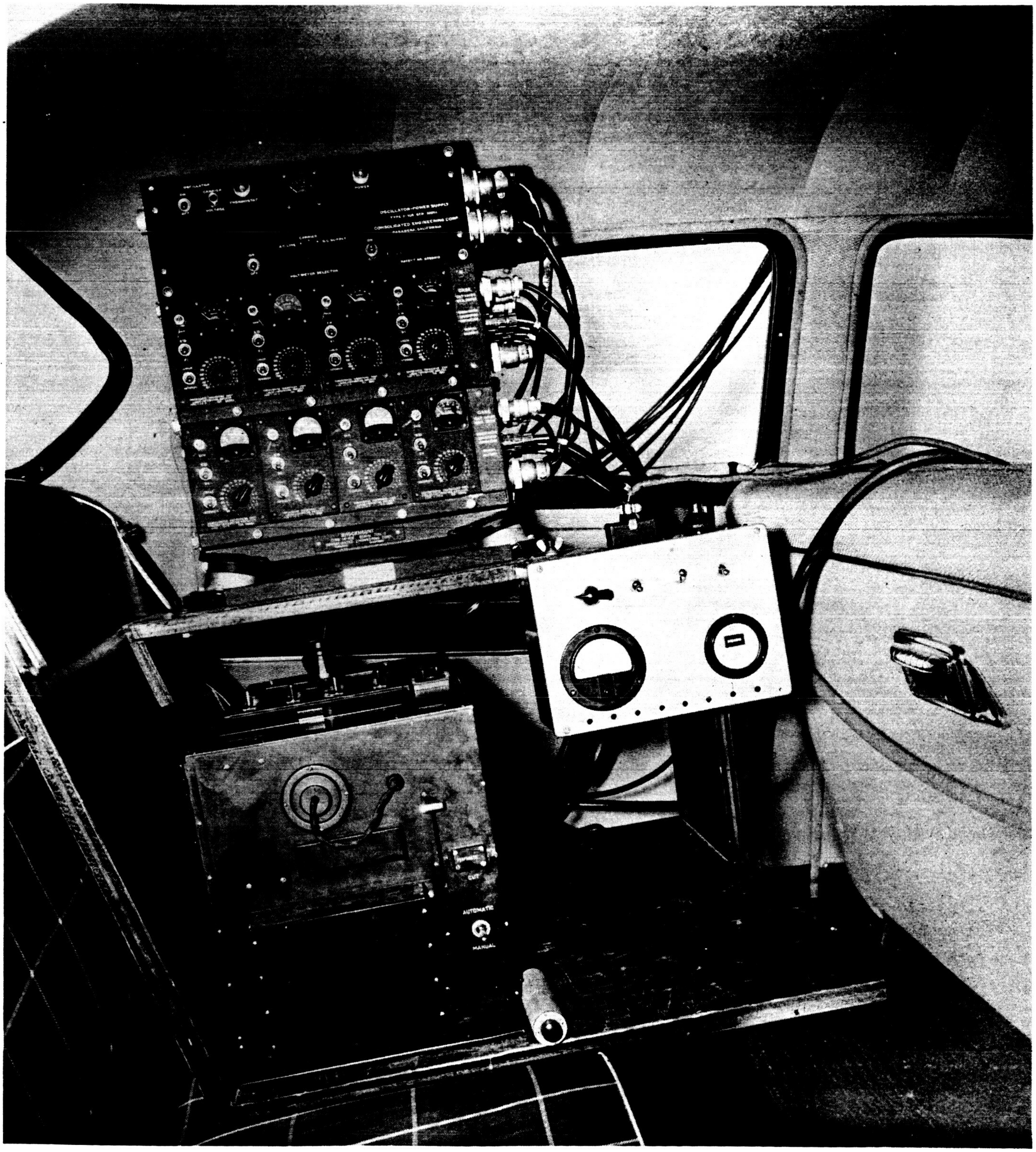

Figure 13 **Multichannel** Amplifiers and Magnetic Oscillograph Used for Vibration and Strain Recording.

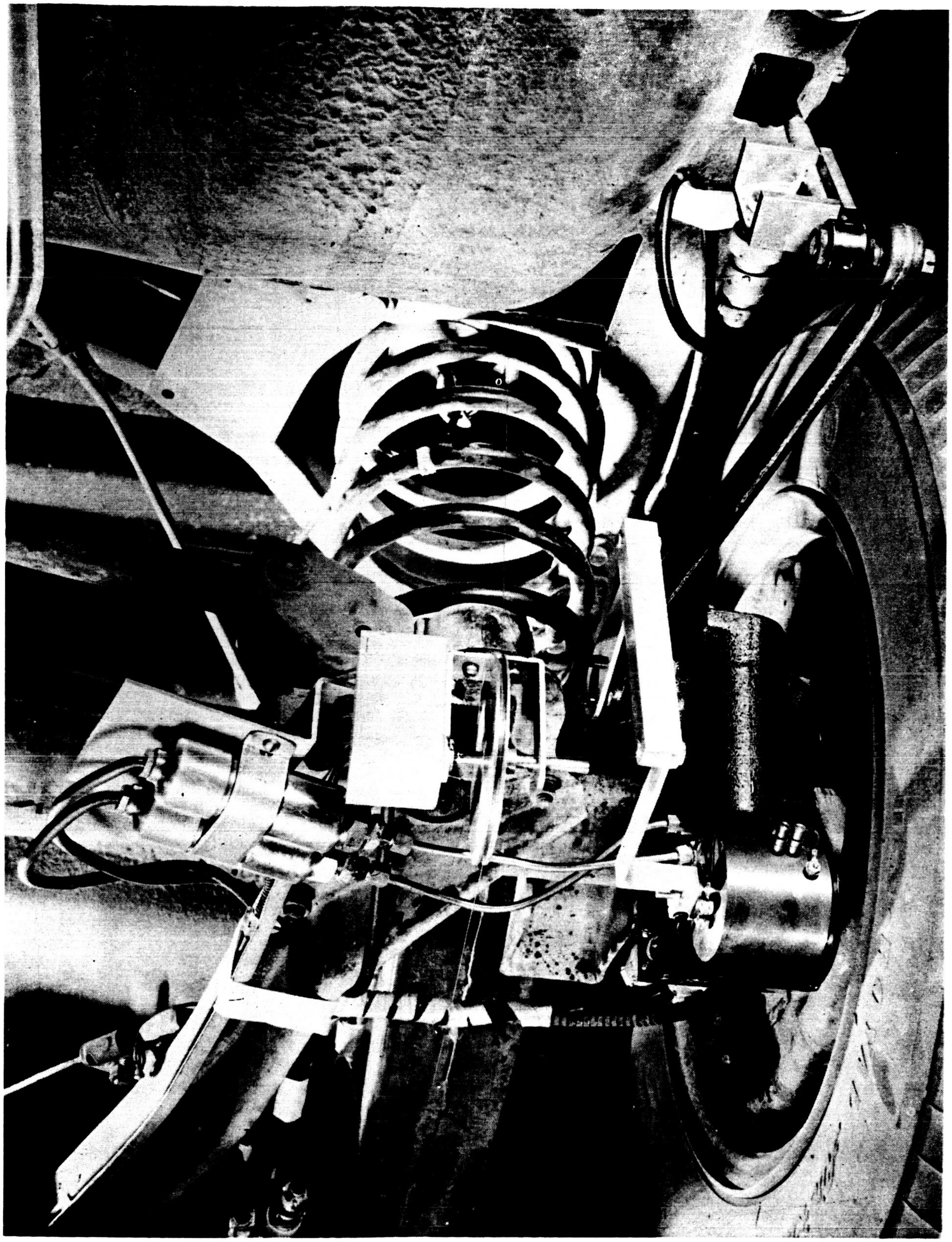

Figure 14 Instrumentation of Rear Shock Absorber. Variables Simultaneously Measured Include Vertical Spring Forces, Axle Displacement, Shock Absorber Forces, Piston Velocity, Piston Displacement, and Hydraulic Pressures.

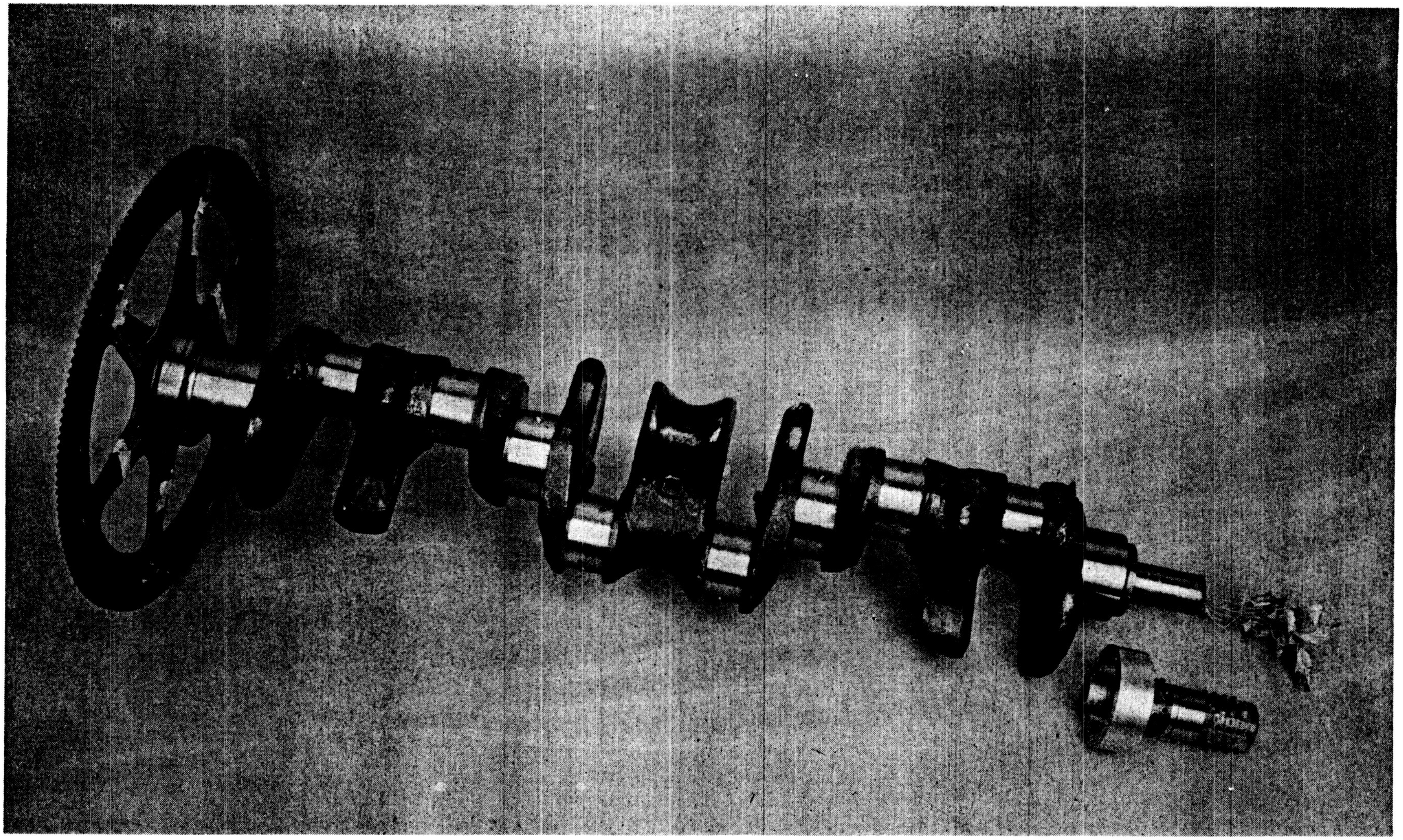

Figure 15 Strain Gages on Flywheel. Wires Threaded Through Crankshaft to Slipring Assembly, Shown Detached from Crankshaft.

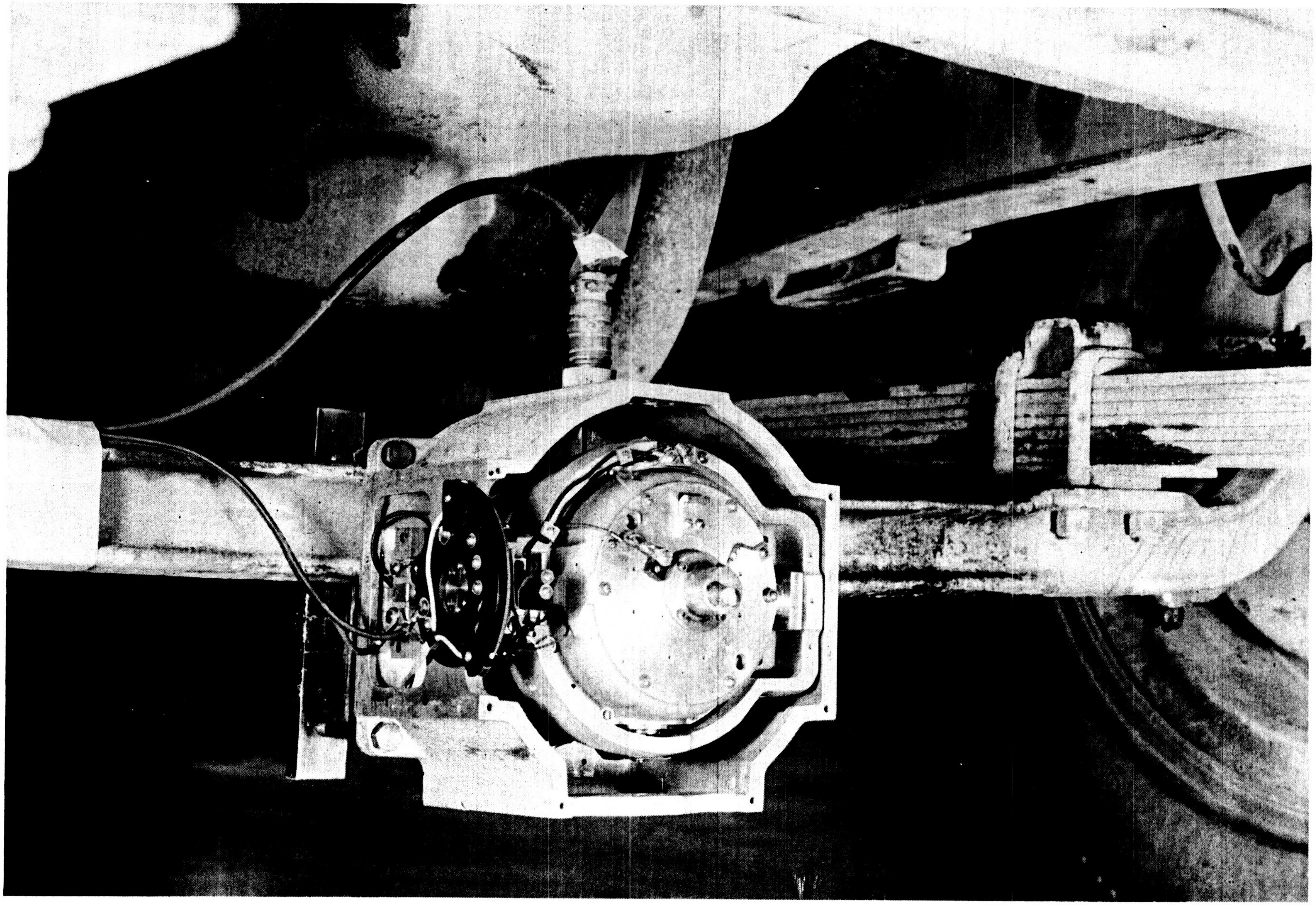

Figure 16 Gyro on Front Axle to Measure Axle Windup During Braking.

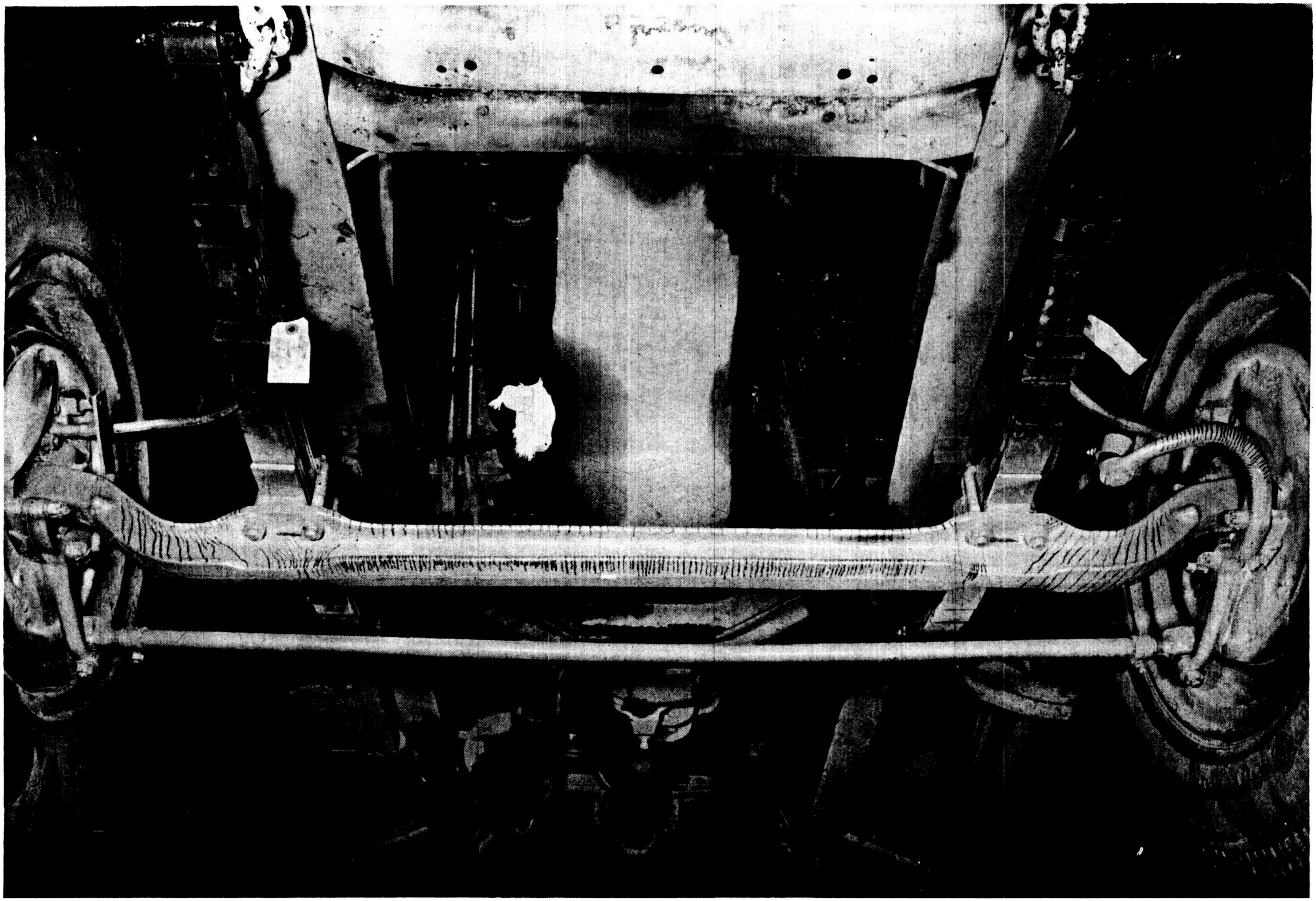

Figure 17 Stresscoat Brittle Lacquer on Front Axle to Determine Bending Stress During Braking. Cracks Have Been Marked with Grease Pencil.

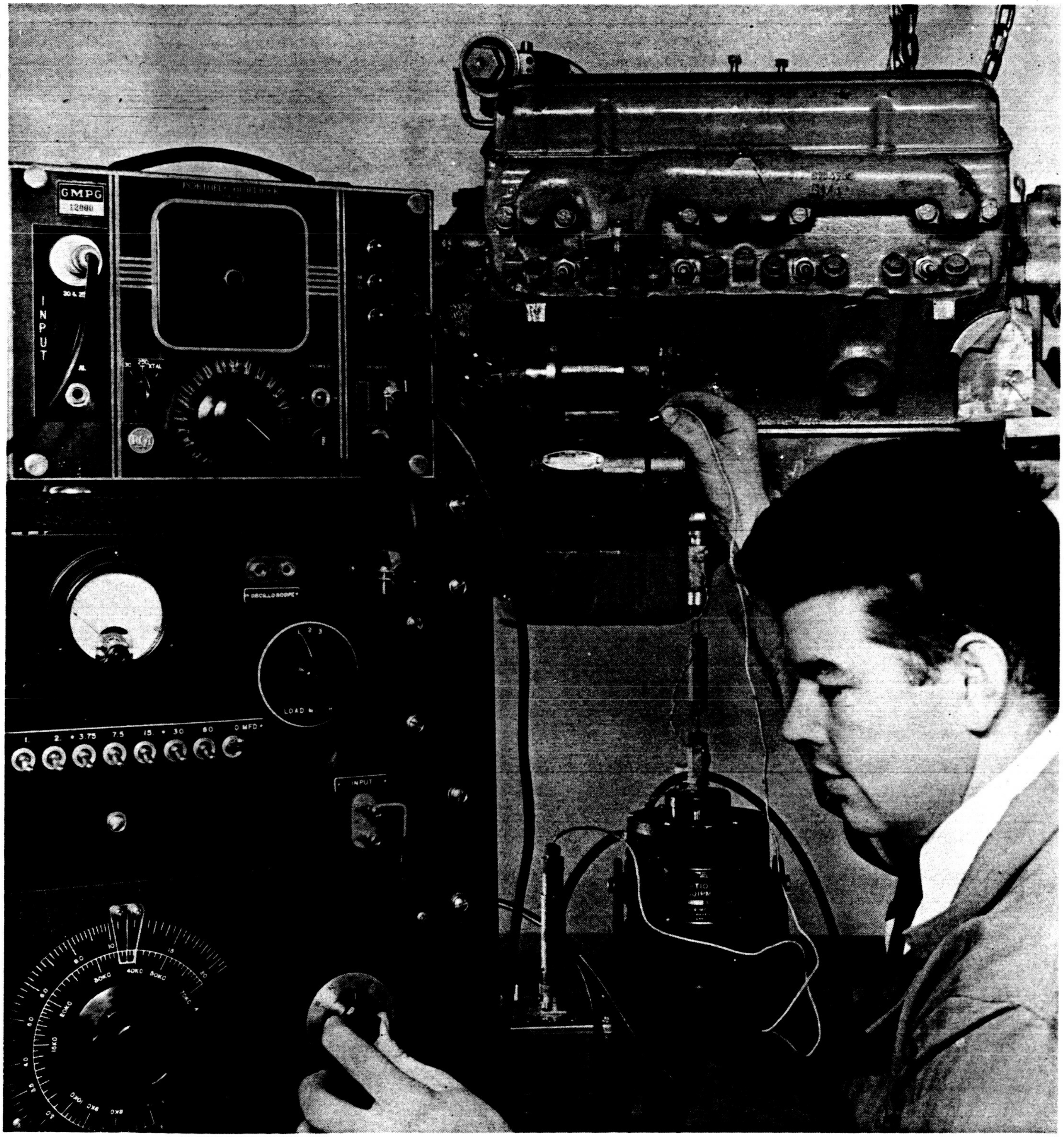

Figure 18 Electrodynamic Shaker Attached to Starting Motor to Determine Its Resonant Frequency and Mode.

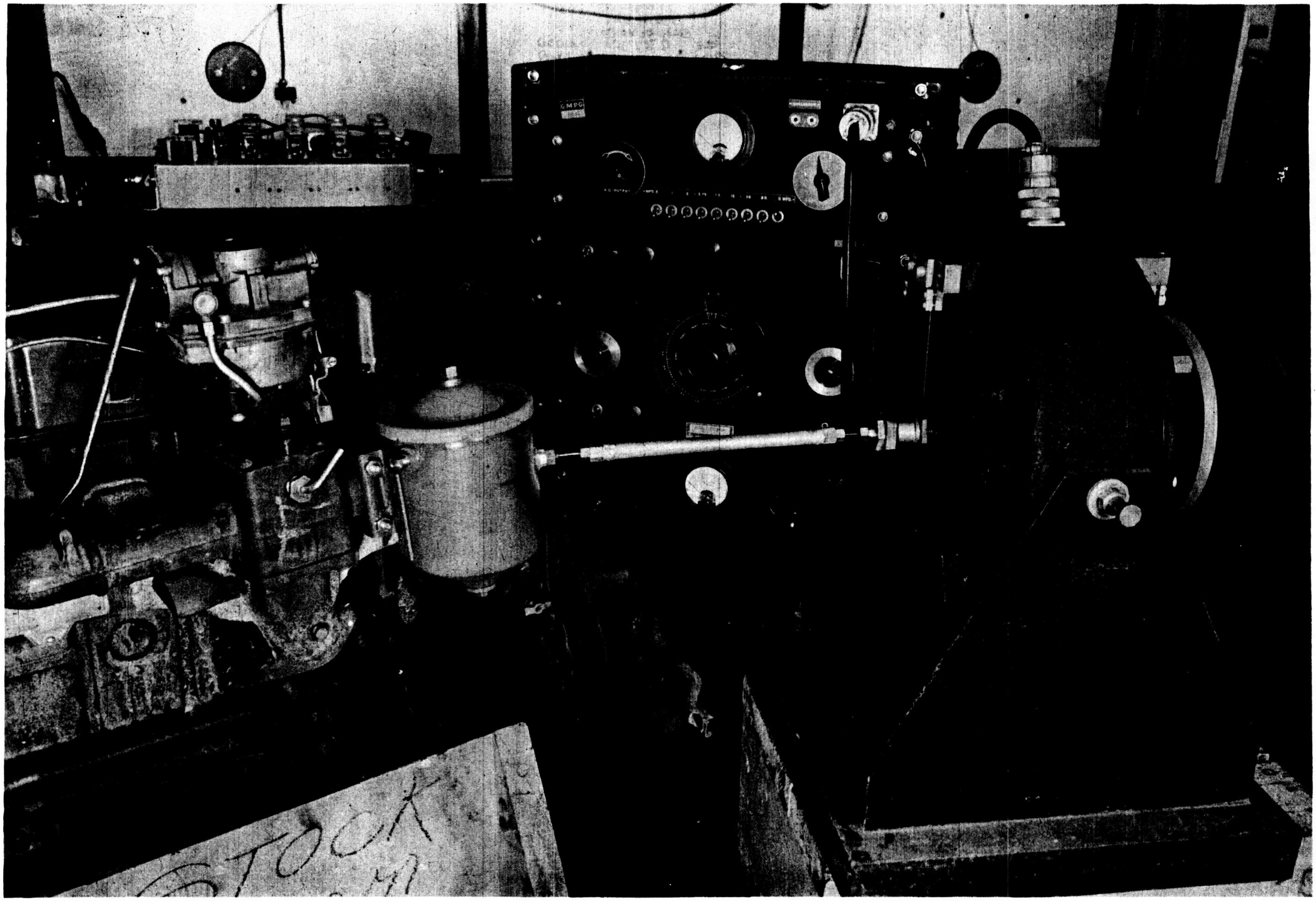

Figure 19 Electrodynamic Shaker Used in Fatigue Studies of Oil Filter.

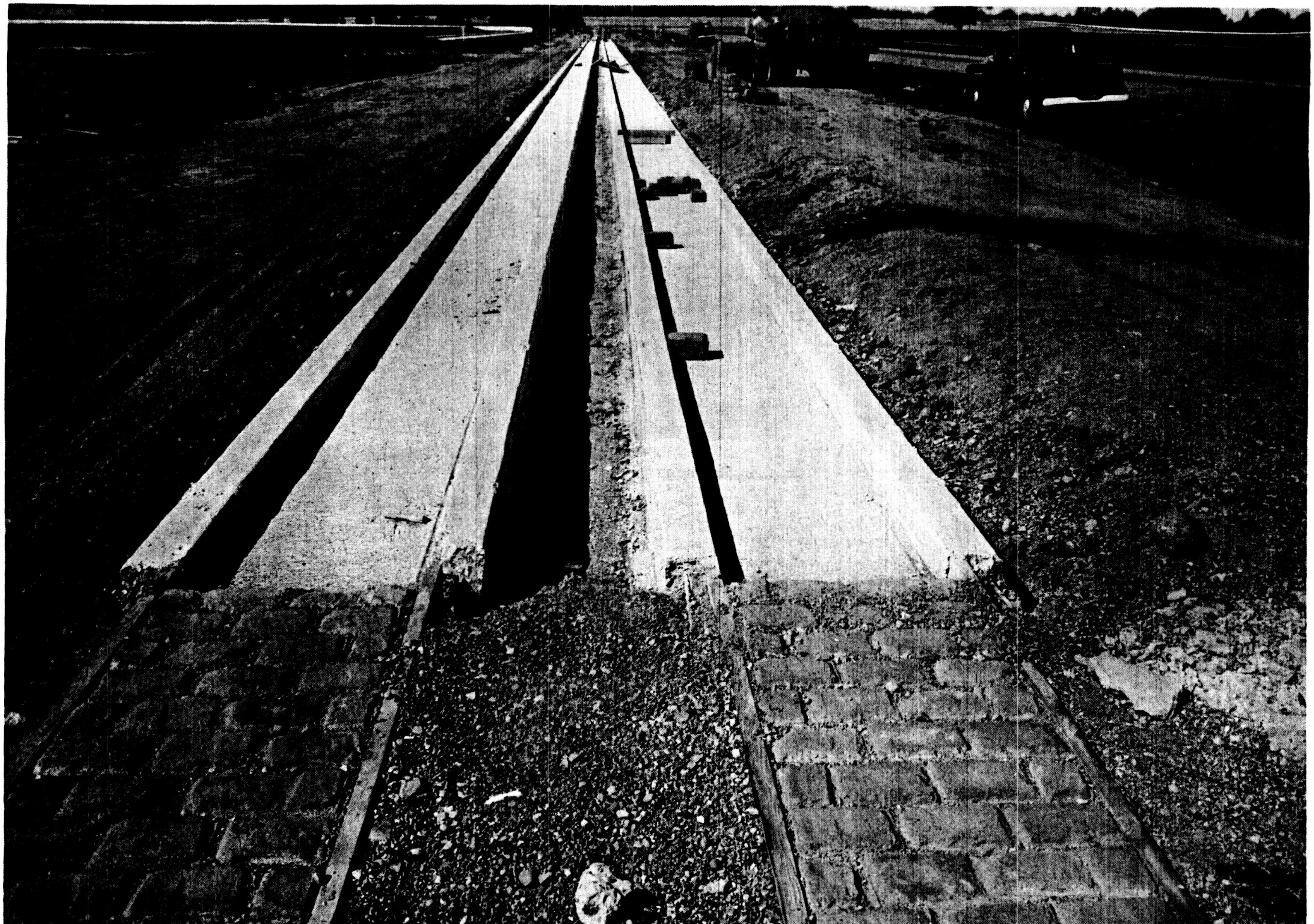

Figure 20 View of German Block Road During Construction.

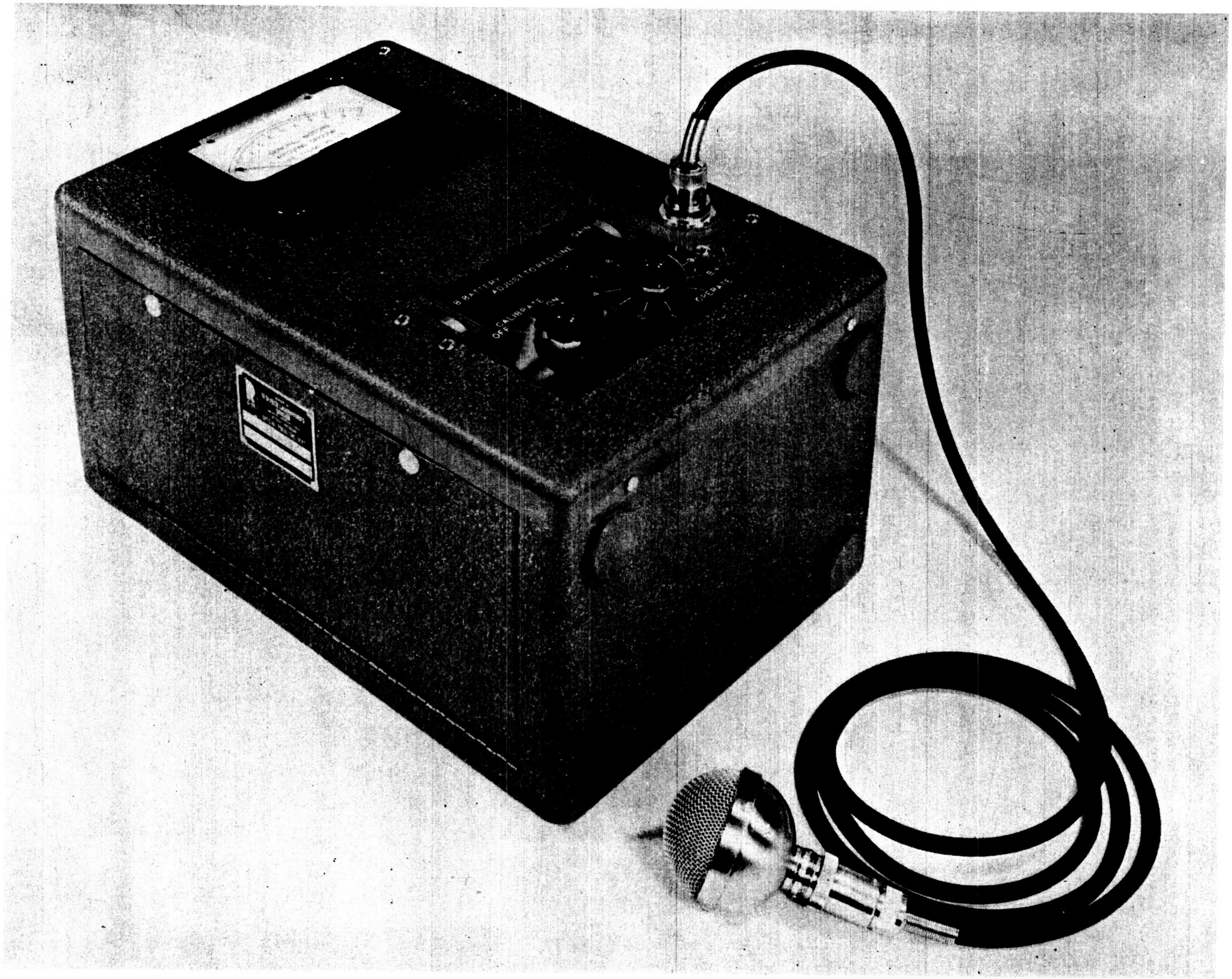

Figure 21 General Motors Tire Thump Meter. This Meter Basically Measures the Beat Envelope of the Sound Pressure which Characterizes this Tire-Excited Disturbance.